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HYPERSONIC RESEARCH ENGINE PROJECT - PHASE IIA
FUEL SYSTEM DEVELOPMENT
FIFTH INTERIM TECHNICAL DATA REPORT
DATA ITEM NO. 55-5.05
20 MARCH THROUGH 19 JUNE 1968
NASA CONTRACT NO. NAS1-6666

AP-68-3995 August 22, 1968

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THE GARRETT CORPORATION
AiResearch Manufacturing Division
LOS ANGELES 45, CALIFORNIA

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PAGE	DATE	PAGE	DATE	PAGE	DATE	PAGE	DATE

FOREWORD

This Interim Technical Data Report is submitted to the NASA Langley Research Center by the AiResearch Manufacturing Company, Los Angeles, California. The document was prepared in accordance with the guidelines established by Paragraph 6.3.3.2 of NASA Statement of Work L-4947-B.

Interim Technical Data Reports are generated on a quarterly basis for major program tasks under the Hypersonic Research Engine Project. Upon completion of a given task effort, a Final Technical Data Report will be submitted.

The document in hand presents a detailed technical discussion of the Fuel System Development for the period of 20 March 1968 through 19 June 1968.

ACKNOWLEDGEMENT

Acknowledgements for significant contributions are extended to the following:

G. L. Chessmore, technical supervision and integration of report contents

J. A. Boppart and I. D. Burke of AiResearch-Phoenix, fuel valve design and development

W. W. English and R. J. Jackson, turbopump analysis

J. L. Sanders, turbopump design

D. K. Yoshikawa, computer program analysis

J. R. Ziolko, systems integration and analysis

CONTENTS

ILLUSTRATIONS

ILLUSTRATIONS (Continued)

APPENDIX A

Symbols	Definition
A_1	supply line upstream orifice, $in.^2$
A_2	supply line exhaust orifice, $in.^2$
A_3	flow area, $in.^2$
A_4	seat area, $in.^2$
A_5	vent orifice, $in.^2$
A_6	actuator chamber flow area, $in.^2$
A_7	metering ball seat area, $in.^2$
A_8	bellows area, $in.^2$
A_9	bellows area (inner PN 393140), $in.^2$
A_{10}	bellows area, (helium PN 395626), $in.^2$
A_{11}	guide flow area, $in.^2$
A_{12}	downstream area, $in.^2$
C_d	discharge coefficient
D	poppet spherical diameter
d	poppet seat diameter
F_a	actuator force
F_p	poppet force
F_T	fluid force
k_8	bellows spring constant
k_9	bellows spring constant
P_1	inlet pressure, psia
P_2	outlet pressure, psia
P_3	actuation pressure, psia

Symbols	Definition
P_4	supply pressure, psia
P_5	vent pressure, psia
P_6	inlet pressure (helium purge), psia
P_7	actuation pressure (helium poppet), psia
R	gas constant
r_P	pressure ratio
r_S	poppet seat radius
r_7	radius, A_7
S_1	poppet stroke, in.
S_2	torque motor wand stroke, in.
S_6	poppet stroke (A_6), in.
S_P	poppet spring
T	temperature
V_1	supply chamber volume, in.^{3}
V_2	bellows volume (between bellows PN 393140), in.^{3}
V_3	actuator chamber volume, in.^{3}
V_4	case volume, in.^{3}

Greek Symbols

γ	specific heat ratio
Δ	denotes difference
$\dot{\omega}$	flow rate

Subscripts

max.	maximum
min.	minimum

1. INTRODUCTION AND SUMMARY

1.1 INTRODUCTION

The Hypersonic Research Engine (HRE) is a regeneratively cooled, hydrogen-fueled ramjet engine intended to be tested on the X-15A-2 aircraft. The fuel system consists both of equipment aboard the X-15A-2 and equipment within the HRE. The equipment aboard the X-15A-2 includes liquid-hydrogen fuel tanks, associated plumbing and pressurization, valves for purging and overboard dump, and other controls required for delivery of the fuel from the storage tanks to the HRE. The equipment within the HRE includes the fuel pump, pump drive, and associated plumbing and control valves to properly meter fuel flow through the fuel injectors and/or overboard fuel dump.

For development purposes, the fuel system is separated into the following major segments:

Fuel system integration

Fuel system valves

Fuel turbopump

Aircraft fuel transfer system

1.2 SUMMARY OF STATUS

The following major efforts were completed during the past quarter for the fuel system:

(a) The computer program for the pressure drop and temperature profile analyses has been completed with all present design revisions incorporated.

(b) Design and fabrication of the breadboard valves has essentially been completed. (All of the valves will be tested in the near future.)

(c) Prototype valve detail drawings were released to manufacturing.

(d) The turbopump design was completed and all parts released to manufacturing. Initial efforts in impeller fabrication using a numerically controlled cutting machine have proved successful, thereby eliminating significant lead times.

Fuel system efforts for the next quarter will include:

(a) Continuation of prototype valve manufacture

(b) Completion of breadboard valve testing

(c) Completion of prototype turbopump manufacture

2. FUEL SYSTEM INTEGRATION

2.1 PROBLEM STATEMENT

An integrated fuel system is to be provided to meet the overall requirements for fuel transfer and regulation to the HRE. To accomplish this, it is necessary to (1) establish the design requirements for the system components, and (2) resolve the interface requirements of the fuel system with the X-15A-2 airplane, the HRE, and the HRE controls.

2.2 BACKGROUND

The fuel system includes the fuel transfer system onboard the X-15A-2 aircraft and the fuel regulation system within the HRE. The integration requirements for the X-15A-2 segment of the fuel system, however, apply only to flow and pressure demands established at the X-15A-2/HRE interface by the fuel control system within the HRE. The analytical design effort for the components in the fuel transfer system is included in the X-15A-2 integrating program. The major portion of analytical effort for fuel system integration will be the establishment of design criteria for the components within the HRE.

The HRE fuel system schematic, Figure 2-1, shows the hydrogen fuel flow paths from the aircraft interface to the combustor fuel injectors. Aboard the aircraft, the fuel flows from the liquid hydrogen storage dewars through a vacuum-jacketed transfer line to the aircraft shutoff and purge valve and quick disconnect at the X-15A-2/HRE interface. From the quick disconnect, the hydrogen flows directly into the pump at a pressure of 50 psia and 40^{0}R. The hydrogen is delivered from the pump discharge at 550 to 1100 psia and approximately 58^{0}R directly into a manifold. At this point, the flow is divided to provide proper cooling of the engine spike, innerbody, outerbody leading edge, and outerbody trailing edge. The flow to each of these cooling passages is regulated by a coolant regulating valve.

As the fuel continues through the cooling paths, it absorbs heat which raises the hydrogen temperature to 650 to 1600^{0}R, depending upon the mode of operation. The hydrogen then enters the fuel plenum, from where it is directed through the fuel control valves into the combustor injector manifolds. A dump valve and lines are provided to dump excess fuel directly overboard, if the required cooling flow exceeds the engine combustor flow demand. Fuel also flows from the outerbody outlet manifolds to the turbine control valve; from here it flows into the turbine and is exhausted directly overboard. The control valve provides the necessary regulation of the turbine to maintain the fuel plenum pressure at 500 to 550 psia.

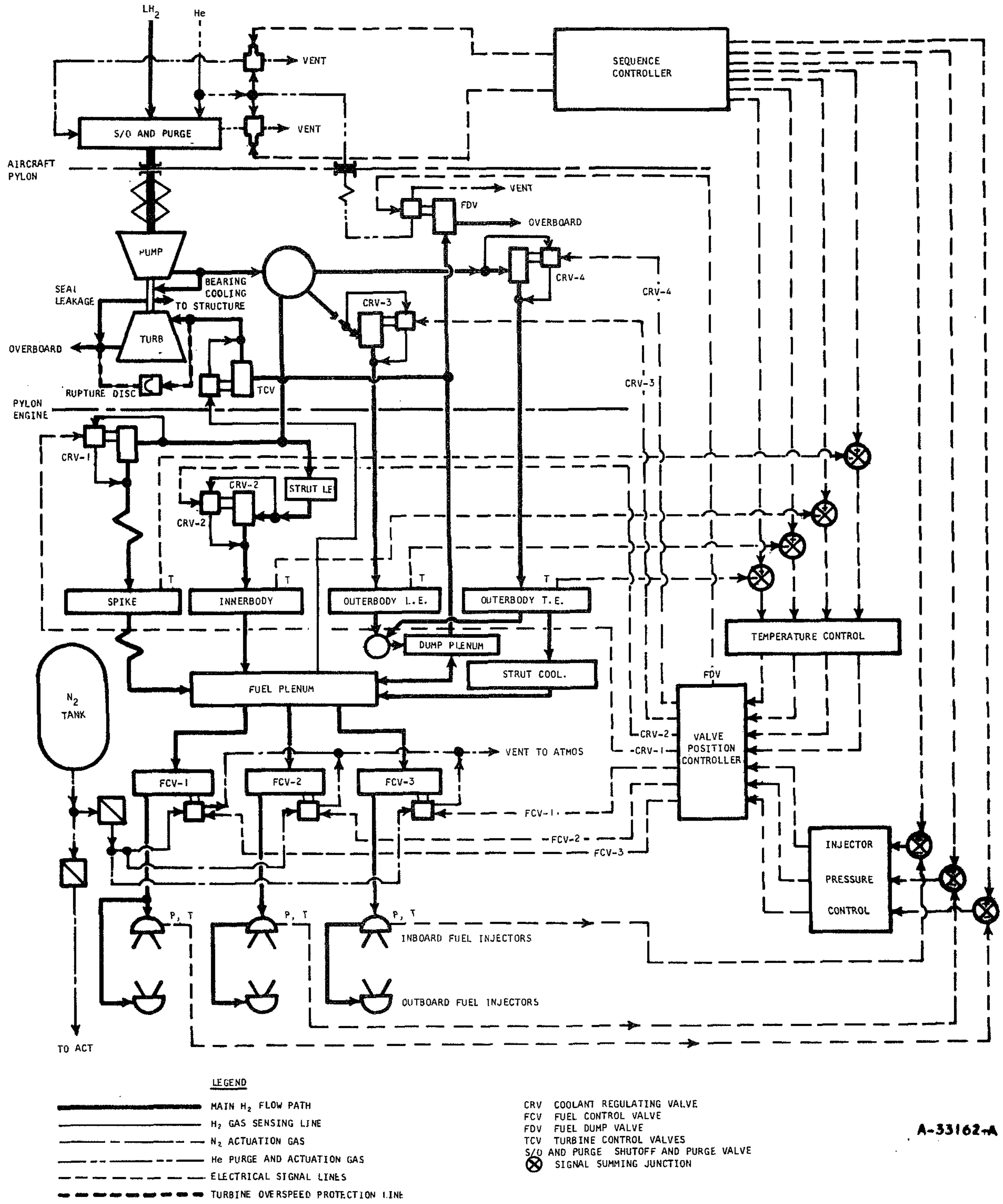


Figure 2-1. HRE Fuel System Schematic

The liquid hydrogen pump also provides the required flow for bearing cooling. The coolant flow is bled from the pump discharge through the bearings in the bearing cavity. From the bearing cavity, some of the flow will leak through the seals into the turbine exhaust, with the remainder available for use in structural cooling.

2.3 OVERALL APPROACH

Fuel system integration will be established by performing various analytical and experimental tasks. The tasks are described in the following paragraphs.

2.3.1 Analytical Approach

The analyses described below will be performed.

(a) Pressure Drop and Temperature Analysis--Establishment of an analysis program for determining pressure drop and temperature profile throughout the fuel flow routes for various flight conditions and engine and cooling flow requirements

(b) Fuel Control Valve Analysis--Determination of operational requirements, flow scheduling, pressure drops allowable, operating pressures and temperatures, dynamic response, etc., for valve design, including leakage requirements, flow area tolerances, and environment

(c) Coolant Regulating Valve Analysis--Determination of operational requirements, flow scheduling, pressure drops allowable, operating pressures and temperatures, dynamic response, etc., for valve design including leakage requirements, flow area tolerances, and environment

(d) Turbine Control Valve Analysis--Determination of operational requirements, flow scheduling, pressure drops allowable, operating pressures and temperatures, dynamic response, etc., for valve design, including leakage requirements, flow area tolerances, and environment

(e) Turbopump Analysis--Determination of operational requirements, operational conditions, pump and turbine flow requirements, pump and turbine power matching capabilities over the flight range, pump cooldown requirements, startup and shutdown transients, and operating environment

(f) Purge and Valve Operating Gas Analysis--(1) Determination of purge requirements for the storage tank and the engine, amount of purge gas available, and the conditions under which it will be available, and (2) determination of the amount of gas required for control valve operation and its availability

(g) Purge and Shutoff Valve Analysis--Determination of operating requirements, flow and pressure drop requirements, operating time, and environment, for valve design

(g) LH$_2$ Tankage and Feedline Analysis--Determination of general operating characteristics of tankage and feedlines, such as storage volume, pressurization gas necessary, pressure decay under flow conditions, g-effects to be considered, environmental effects to be considered, feedline pressure drop, feedline heat transfer rates, and schedule of pump inlet conditions

(h) Failure Mode Analysis--Determination of safety and reliability requirements and preparation of a component-by-component analysis to determine system ability to meet the requirements

(i) Analog Computer Analysis--Dynamic analysis of the fuel system (turbine control loop and fuel control loop) to help define system configuration and ability of system and components to function properly in all required flight regimes

The details of specific design problems and the analytical solutions are given in Section 2.4.

2.3.2 Experimental Approach

One prototype system, consisting of refurbished prototype components, will be used for fuel system development testing. The test system will include all fuel control and coolant regulating valves, the purge and shutoff valve, the turbopump and its control valve, and simulated plumbing and restrictions in the fuel system loop. Generally, system development testing will be performed under steady-state and transient heat transfer conditions to provide assurance that the fuel system is capable of performing under all modes of operation, including combined cooling and combustion flow. Specific tests and objectives are given below.

Functional checkout of all system and test setup valves and controls to verify that general operational and safety requirements are met

Fuel flow calibration for the engine cooling mode and combined cooling and combustion modes

Steady-state-point operational checkout with transient step inputs to obtain dynamic performance data

Startup and shutdown operational checkout to evaluate transient times and dynamic stability

Purge system operational checkout to ascertain response time and flow rates

Complete performance and operational sequencing checkouts from startup to shutdown to evaluate compatibility with fuel system controls

Failure mode checkout to determine compatibility with safety requirements

2.4 ANALYTICAL EFFORT

2.4.1 System Analysis

The computer program for calculation of fuel system performance data, including pressure drop and temperature profiles, has been revised to incorporate all physical changes in the routing of the fuel loop.

A complete description of the revised program is contained in a separate report, AP-68-4003.

2.4.2 Reliability Effort

Reliability engineering support of the fuel system design effort has been primarily concerned with hardware design analyses. Failure Modes, Effects, and Criticality Analyses (Data Item 21) and Design Checklists for Maintainability and the Elimination of Human Induced Failures (Data Item 22) for the redesigned fuel system valves were transmitted to the NASA. The Failure Mode, Effects, and Criticality Analysis for the redesigned hydrogen turbopump has also been submitted. Surveillance over the fuel system testing effort is continuing.

2.4.3 Analog Computer Analysis

The Control System Development Fifth Interim Technical Data Report (Data Item No. 55-6.05) contains a description of the fuel system analog simulations which include the injector system and the temperature control. The injector simulation covers at present only the first injector. The components which are included in this simulation are the fuel lines from the main fuel plenum, the injector manifolds, the fuel control valve, and the digital computer. There are two temperature control simulations: (1) the main coolant system which includes the lines, manifolds, coolant regulating valves, and heat exchangers from the discharge of the turbopump to the main fuel plenum, and (2) the turbopump system, which includes the turbine control valve and the LH_2 turbopump.

2.5 TEST ACTIVITY

Design of the test area for the turbopump and system testing with hydrogen has been initiated. The facility will include a 14,000-gal capacity storage system with an expulsion capability of 3.0 lb per sec of liquid hydrogen delivered at 60 psia, a high capacity cryogenic vaporizer to vaporize the pump discharge flow for venting purposes, and a high temperature gaseous hydrogen heater to simulate the engine heat energy input. Insofar as is possible, flow paths, volumes, etc. will duplicate those of the HRE; however, the physical size of the vaporizer, heater, and interconnecting plumbing will not allow duplication of engine transient response inputs to any great extent.

3. FUEL CONTROL VALVES

3.1 PROBLEM STATEMENT

A set of valves is to be provided to regulate the hydrogen flow through the HRE to satisfy the fuel requirements for structural cooling and engine operation. This set of valves consists of four cooling flow regulating valves, three fuel control valves (one for each of the three engine combustor injection stations), a dump valve for overboard dumping of excess fuel requiredfor cooling (but not engine operation), a throttling valve for control of the turbopump, and a valve that combines the functions of fuel shutoff and engine purging.

These valves and their assigned designations are:

Fuel Control Valve, FCV-1	PN 393090
Fuel Control Valve, FCV-2	PN 393088
Fuel Control Valve, FCV-3	PN 393090
Coolant Regulating Valve, CRV-1	PN 393142
Coolant Regulating Valve, CRV-2	PN 393142
Coolant Regulating Valve, CRV-3	PN 393142
Coolant Regulating Valve, CRV-4	PN 393142
Fuel Dump Valve, FDV	PN 393140
Turbine Control Valve, TCV	PN 393094
Purge and Shutoff Valve	PN 395626

3.2 BACKGROUND

The requirements for the fuel system valves are dictated by engine performance, structural cooling, and fuel flow regulation requirements necessary for engine startup, operation, and shutdown.

Specific factors that influence the design of the various valves are the required fuel flow, fuel pressure, flow accuracy, flow dynamic response, hydrogen gas temperature, structural envelope and mounting requirements, engine environmental conditions, and reliability and safety considerations.

The hydrogen fuel is stored as a liquid at low pressure onboard the X-15A-2 vehicle. It is transferred to the HRE through vacuum-jacketed lines. At the X-15A-2/HRE interface, the hydrogen fuel flows through the shutoff and purge valve to the turbopump, where it is boosted to the high pressure required to provide flow through the structural cooling jackets, the engine fuel control valves, and the fuel ejectors to the engine. Both pressure and flow are determined by the design characteristics of the structural cooling jackets and fuel ejectors required to obtain the desired engine performance.

3.3 OVERALL APPROACH

The overall aim of the fuel system valve development program is to establish design configurations that will satisfy performance requirements, with regard to providing the required flows, pressures, etc.; dynamic performance, with regard to response characteristics for total engine control system performance; and the safety and reliability requirements, with regard to mission and pilot safety.

This objective is to be achieved by an analytical effort to establish preliminary design configurations that will satisfy performance criteria based upon established fuel system concepts. This effort will be followed by a system analysis of the valves, to ensure that proper dynamic performance is obtained, and then by a failure modes and effects analysis to ensure reliability goals. The analytical studies will then be verified by fabrication and testing of breadboard and prototype models prior to manufacture of flight hardware.

3.4 ANALYTICAL EFFORT

3.4.1 Design Summaries

Design summaries for all valve sizing and performance data are completed except for valve 393142. The design summary for valve 393142 will be completed when downstream requirements have been determined from computer runs. The completed analyses are contained in Appendix A.

3.4.2 Stress Calculation

Stress calculations were completed for the fuel system valves as designed. Minor changes were recommended to increase the minimum thickness of the hot gas valves actuator covers.

The analyses indicated that, for the most part, the stress levels are conservative, and were based on the formulas presented in MIL-Handbook -5A, "Strength of Metal Aircraft Elements," and Roark's "Formulas for Stress and Strain."

The calculations will be submitted in a separate report.

3.5 DESIGN EFFORT

3.5.1 Breadboard Valves

The design and fabrication of all breadboard valves is essentially complete. All units, except for PN's 393142 and 393140, are in test. The fabrication of these two units was delayed due to difficulties encountered in heat treatment of the bellows. New bellows have been received and assembly of the units is expected in the immediate future.

3.5.2 Prototype Valves

Prototype valve detail drawings have been released and manufacturing is continuing on schedule relative to the required completion dates.

3.6 TEST ACTIVITY

Initial testing of the breadboard valves has included:

(a) Helium leakage tests on both the poppets and the housing on PN 395626

(b) Solenoid and pneumatic actuator operation for both the hydrogen and helium flow sides of PN 395626

(c) Flow tests on the helium side of PN 395626

(d) Servo controller operational tests on PN's 393088 and 393090

After initial testing, the valves will be fitted with test adapters for mounting in test stands for hydrogen and hot-air flow testing.

3.7 FUTURE ACTION

The effort during the next quarter will be to continue the prototype manufacturing effort and complete testing of the breadboard units.

4. FUEL TURBOPUMP

4.1 PROBLEM STATEMENT

A fuel turbopump is to be provided for pressurization of the liquid hydrogen from storage pressure to the high-pressure level necessary to obtain the required flow rates through the structural cooling paths and engine fuel injectors.

4.2 BACKGROUND

The turbopump discharge pressure and flow design criteria are dictated by the engine performance and structural cooling requirements. Analysis of these requirements show the necessity for a high-pressure pumping system. Several high-pressure systems besides the turbopump were considered in earlier analyses but were eliminated mainly on the basis of excessive weight. The unique feature of the turbopump is that the turboexpander drive unit uses the heat energy absorbed by the hydrogen coolant, thus eliminating the necessity for an external energy source, while still allowing utilization of the hydrogen for combustion.

4.3 OVERALL APPROACH

The overall approach to the turbopump design is to establish preliminary concepts from previous designs, complete a detailed analytical program to size the turbine and pump units, and then construct prototype subassembly units for experimental verification of the design parameters.

4.4 ANALYTICAL EFFORT

The critical speed analysis has been completed. The analysis indicates that no critical speeds are expected within the normal operating range of 55,000 to 85,000 rpm. The analysis holds true throughout and well beyond the anticipated range of bearing spring stiffness rates. The unit will pass through the first and second critical speeds; however, these will both be solid body modes and no problem is expected. The third critical speed, which is a flexural mode, is well above any possible operating speed. The analysis was submitted as Data Item 54.01, "Stress and Structural Calculations for Hydrogen Fuel Turbopump Analysis."

4.5 DESIGN EFFORT

4.5.1 Design

The layout of the turbopump was completed and approved. This design incorporates the labyrinth-seal concept. Detail drawings of all parts were completed, and released for fabrication. A weight analysis of the final design has been completed and the calculated weight of the design is 30 lb.

4.5.2 Fabrication

Fabrication of two sets of all detail parts was initiated. The impeller blading is being cut on a 5 axis, numerically controlled machine. A sample blade section has been completed and is currently undergoing inspection to confirm blade configuration. The turbine wheel blades will be fabricated utilizing the Anocut process. These blades will require minor hand finishing at the inlet tips to meet the requirements of the HRE designs. Bearings have been received. All other parts are in process.

4.6 TEST ACTIVITY

The subcomponent testing of the turbopump components has been scheduled to obtain information in regard to bearing design, leakage rates through the labyrinth seals, and evaluation of various metal seals to be used on the turbopump. Two test rigs have been designed; one for the bearing tests, and one for the seal tests. They were described in the previous fuel system development TDR, Data Item 55-5.04.

4.6.1 Bearing Test Rig

The bearing test rig has been completed. The unit has been forwarded to the Boron Test Facility, and is being set up. A photograph of the bearing test rig is shown in Figure 4-1.

4.6.2 Seal Test Rig

Testing of labyrinth seal leakage rates using liquid nitrogen has been completed. The results correlate closely with current labyrinth seal theory. Results are presented in Figure 4-2.

4.6.2.1 Test Fixture

The radial labyrinth configuration chosen was a reversed, straight-through labyrinth seal. Two sets of lands were machined on the shaft side. Each set of lands contained five teeth and had a relative chamber width of 15. The chamber depth is not as significant, so a chamber width to depth ratio of 1.5 was chosen.

According to available design literature, rotation and eccentricity have no significant effect on leakage. Therefore, testing was conducted on non-rotating seals.

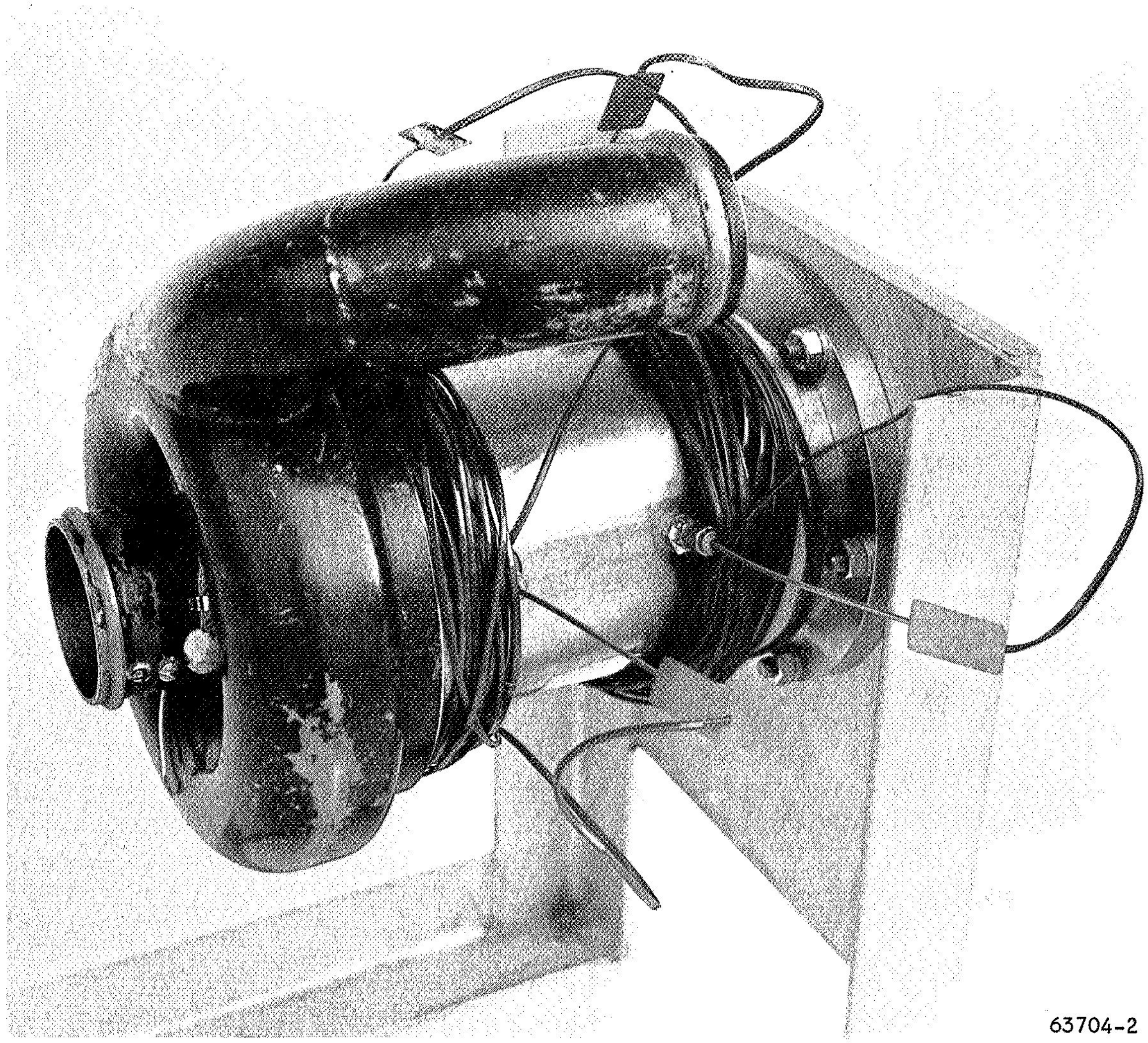


Figure 4-1. Bearing Test Rig

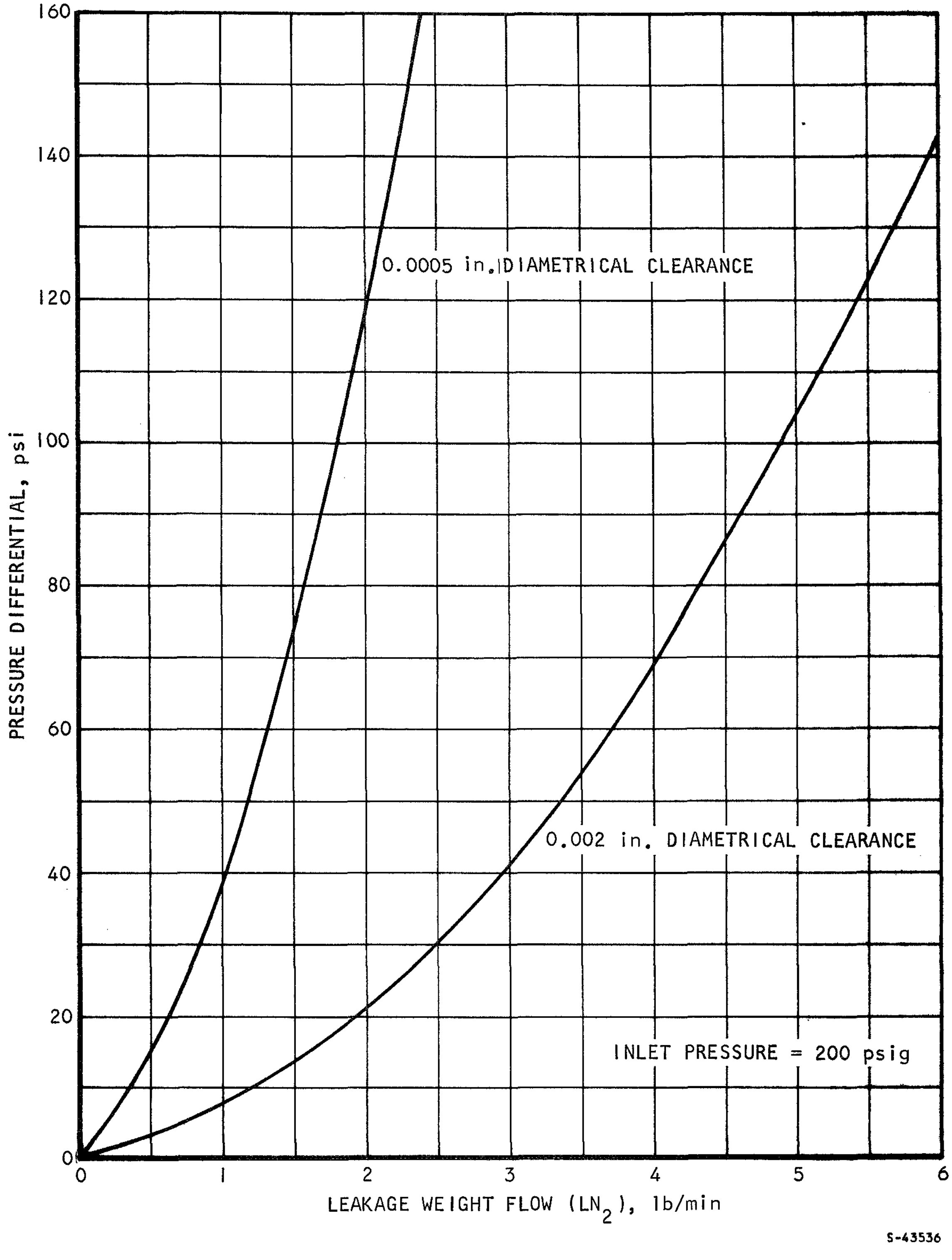


Figure 4-2. Labyrinth Seal Test, Flow (LN_2) vs Pressure Drop

Diametral clearances of 0.0005 and 0.002 in. were selected for investigation because of the small molecular size of hydrogen and to avoid rubbing of the teeth. The seal test rig is shown in Figures 4-3, 4-4, and 4-5. Figures 4-3 and 4-4 show the inlet and outlet ends of an assembled unit, and Figure 4-5 shows the seal configuration of a disassembled unit.

4.6.2.2 Test Procedure

The test setup is as shown in Figure 4-6.

The test procedure is as follows:

(a) Test each seal configuration with LN_2.

1. With a 200 psig inlet, maintain a discharge pressure of 10 psig for approximately 3 hr. Record any degradation of flowrate.

2. If the flow degraded in step 1, report the flow rates to the cognizant engineer.

3. If flow degradation does not occur, measure the static pressure drops across the fixture for various flowrates.

(b) Repeat step (a) using LH_2 and with an inlet pressure of 100 psig.

(c) For subsequent seal configurations, test with LH_2 per step (b)-3 above.

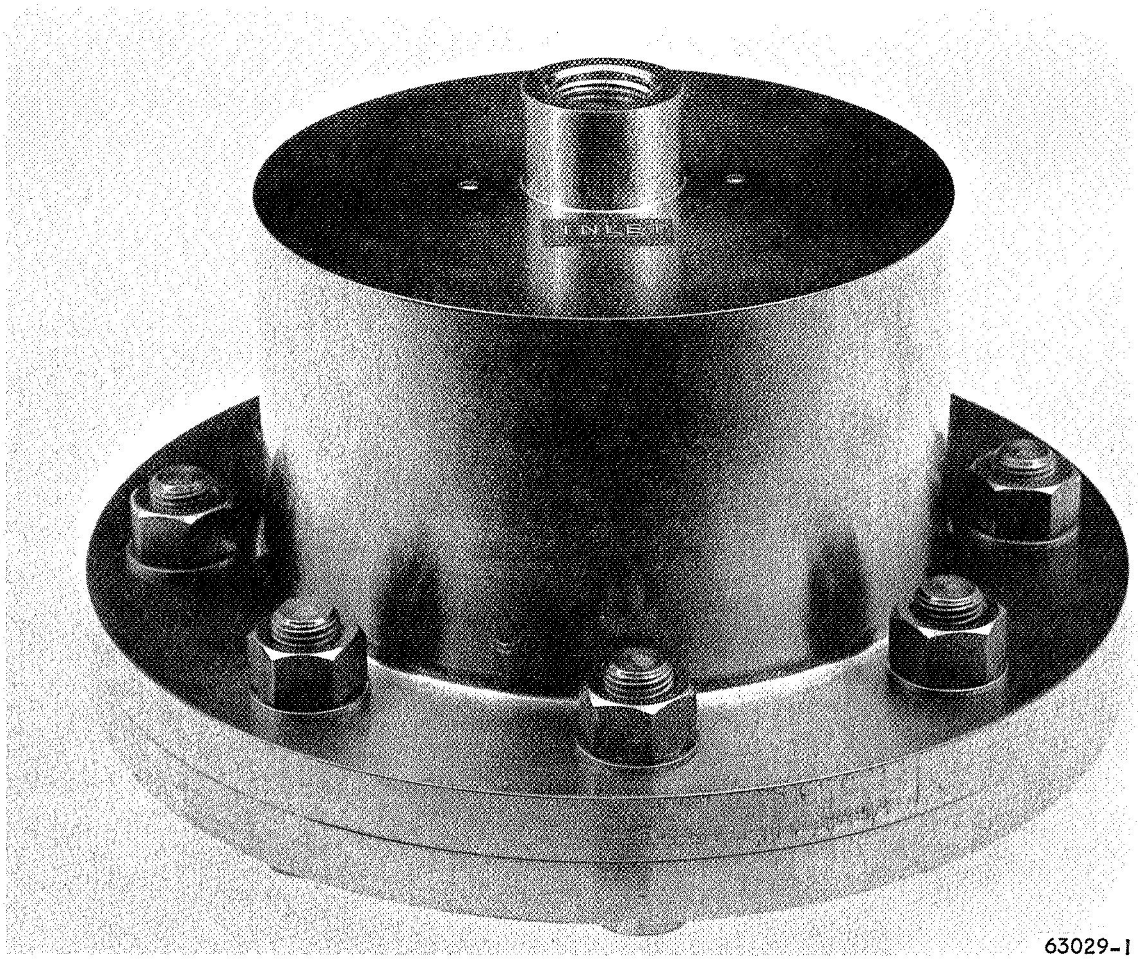


Figure 4-3. Labyrinth Seal Test Rig, Inlet End

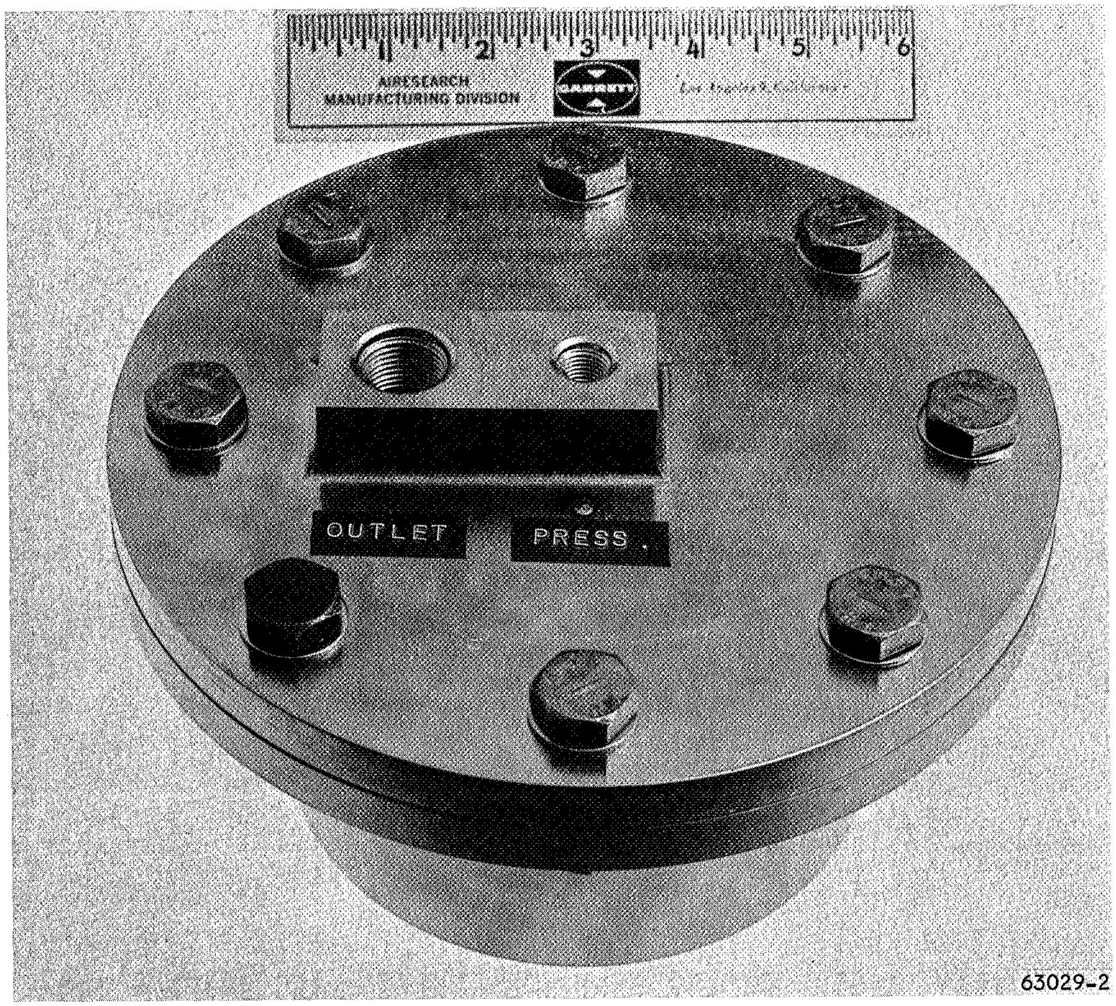


Figure 4-4. Labyrinth Seal Test Rig, Outlet End

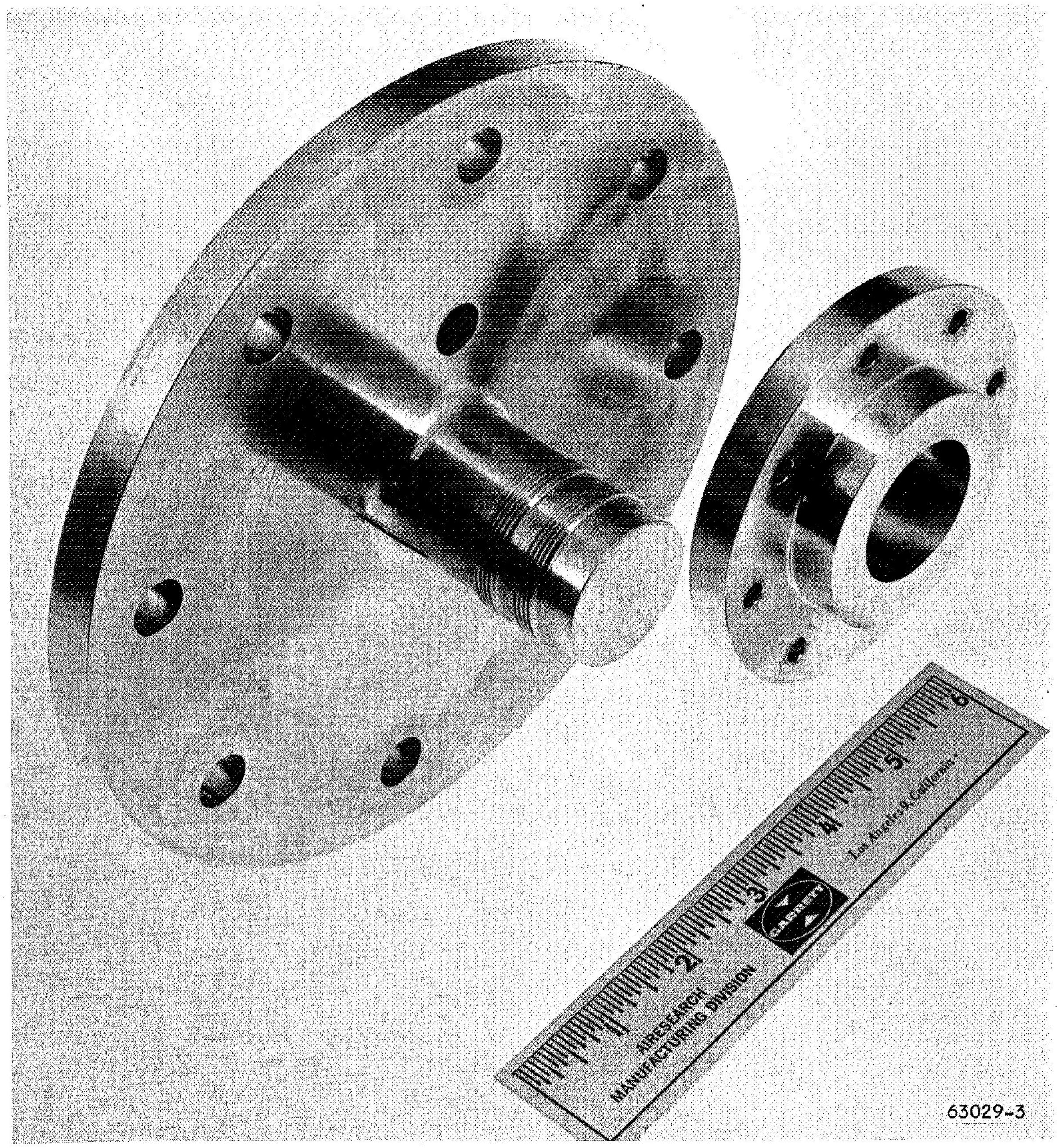


Figure 4-5. Labyrinth Seal Test Rig, Seal Configuration

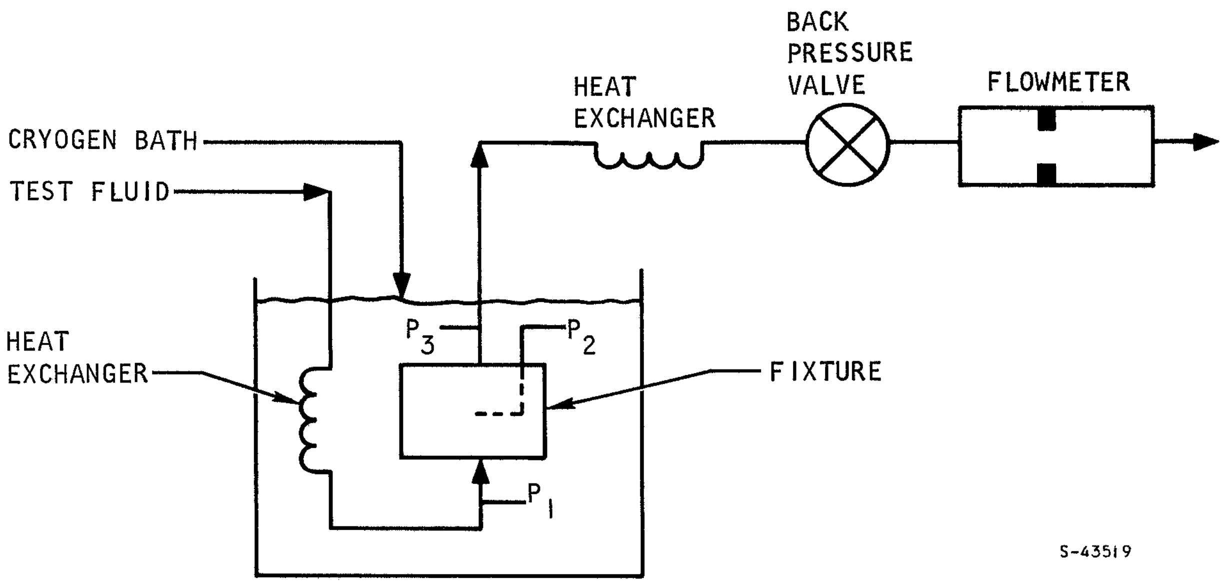


Figure 4-6. Seal Test Rig Test Setup

APPENDIX A

HRE FUEL VALVE ANALYSIS

APPENDIX A

HRE FUEL VALVE ANALYSIS

An analysis of the fuel system valves, PN's 393090, 393088, 393094, 393140, and 395616, was conducted to determine the sizing and operating characteristics in accordance with the requirements presented in the Fuel System Valves Statement of Work, AiResearch Report No. 67-1897. The design calculations are presented below. Schematics for the valves are presented in Figures A-1 through A-5.

FLOW AREA

Flow area sizing for all valves, except PN 395626, was determined from the following equation (the definitions for all of the symbols used in this appendix can be found in the table of symbols at the front of this report):

$$A_4 = \frac{\dot{\omega}\sqrt{RT_1}}{8.02C_dP_1\sqrt{\frac{\gamma}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}} \qquad \text{(A-1)}$$

where

Parameter \ Valve PN	393088	393090*	393094	393140
$\dot{\omega}$, lb/sec	1.0	1.6	0.25	2.75
R, ft-lb/lb-^oR	767	767	767	767
T_1, ^oR	1600	1600	1600	1400
C_d	0.6	0.6	0.6	0.6
P_1, psia	500	500	525	300
P_2, psia	450	450	505	$P_1/P_2 > 1.8$
γ, C_p/C_v	1.4	1.4	1.4	1.4
A_4, sq in.	1.539	2.46	0.568	4.08
d, in.	1.40	1.77	0.850	2.28

*Breadboard configuration only (final configuration flow = 2.0 lb/sec)

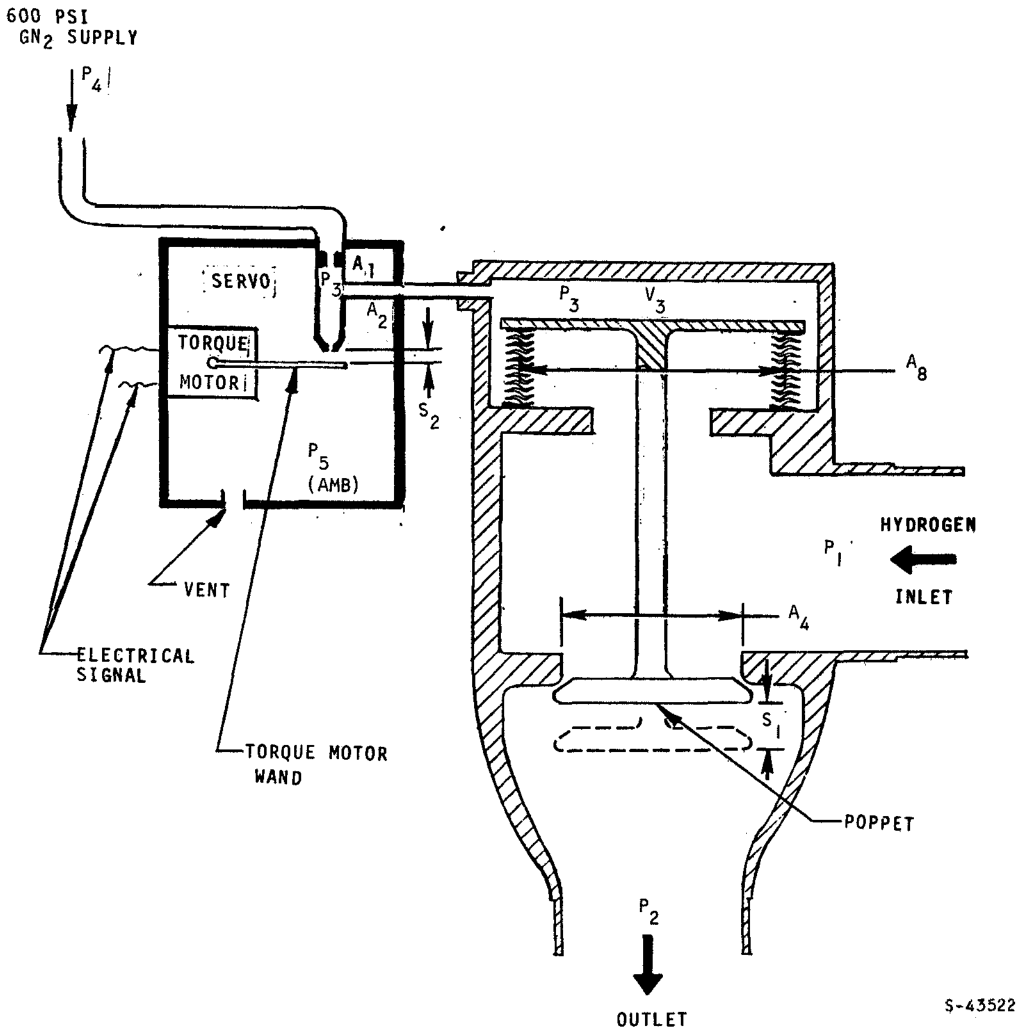


Figure A-1. Fuel Control Valve Schematic (PN 393090)

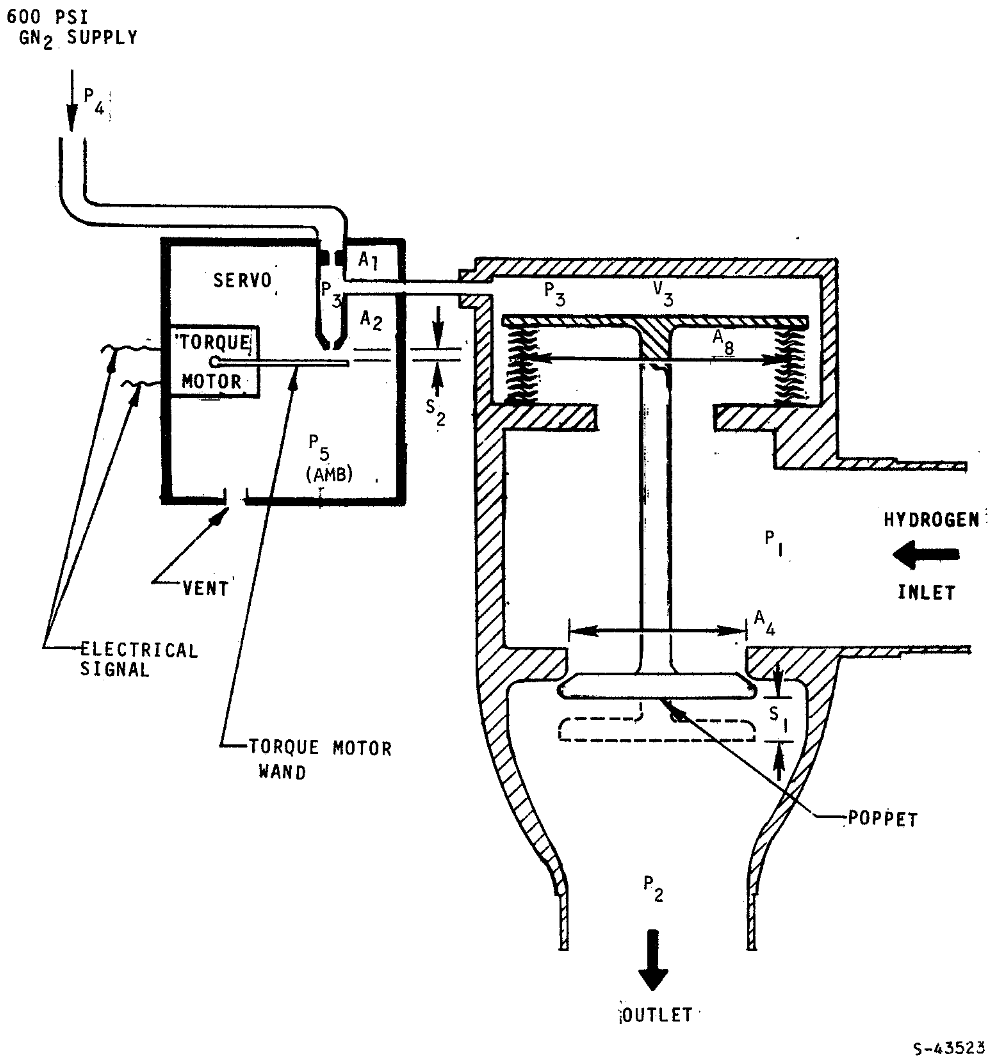


Figure A-2. Fuel Control Valve Schematic (PN 393088)

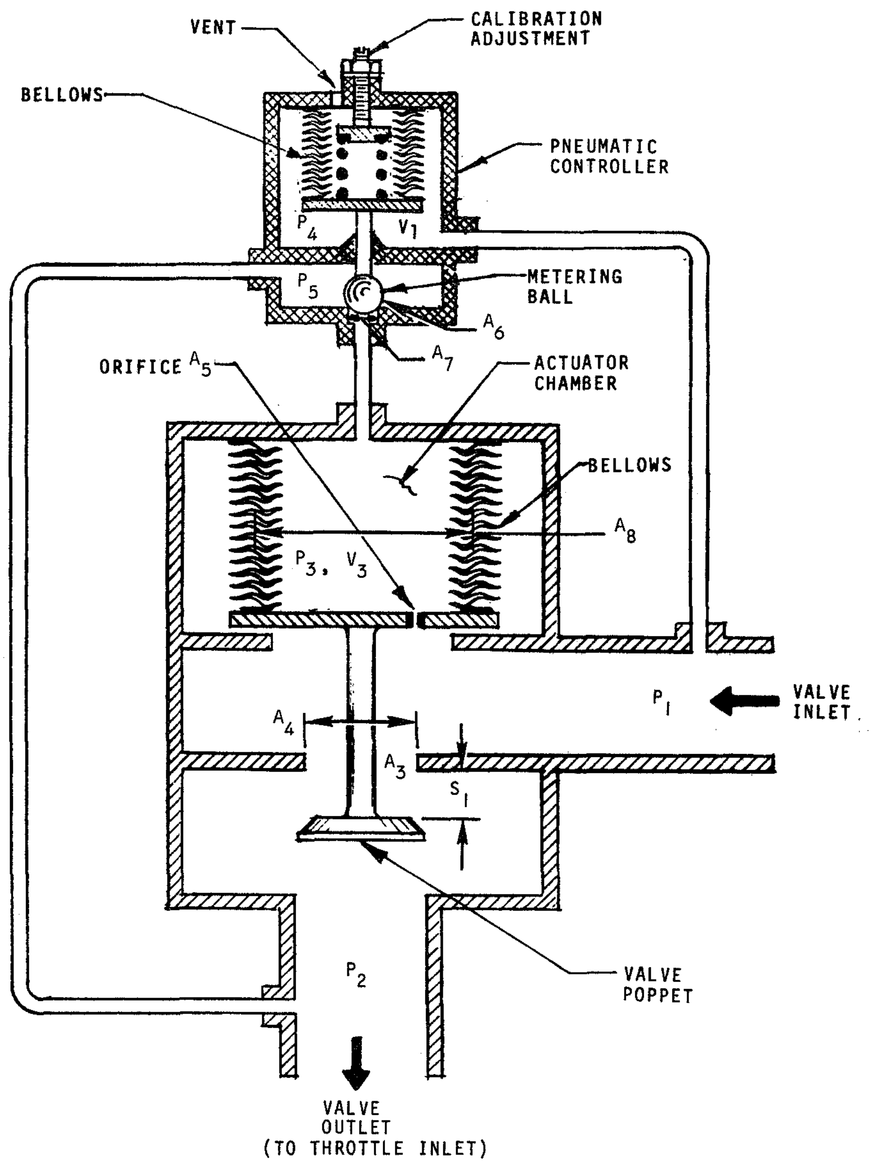


Figure A-3. Turbine Control Valve Schematic (PN 393094)

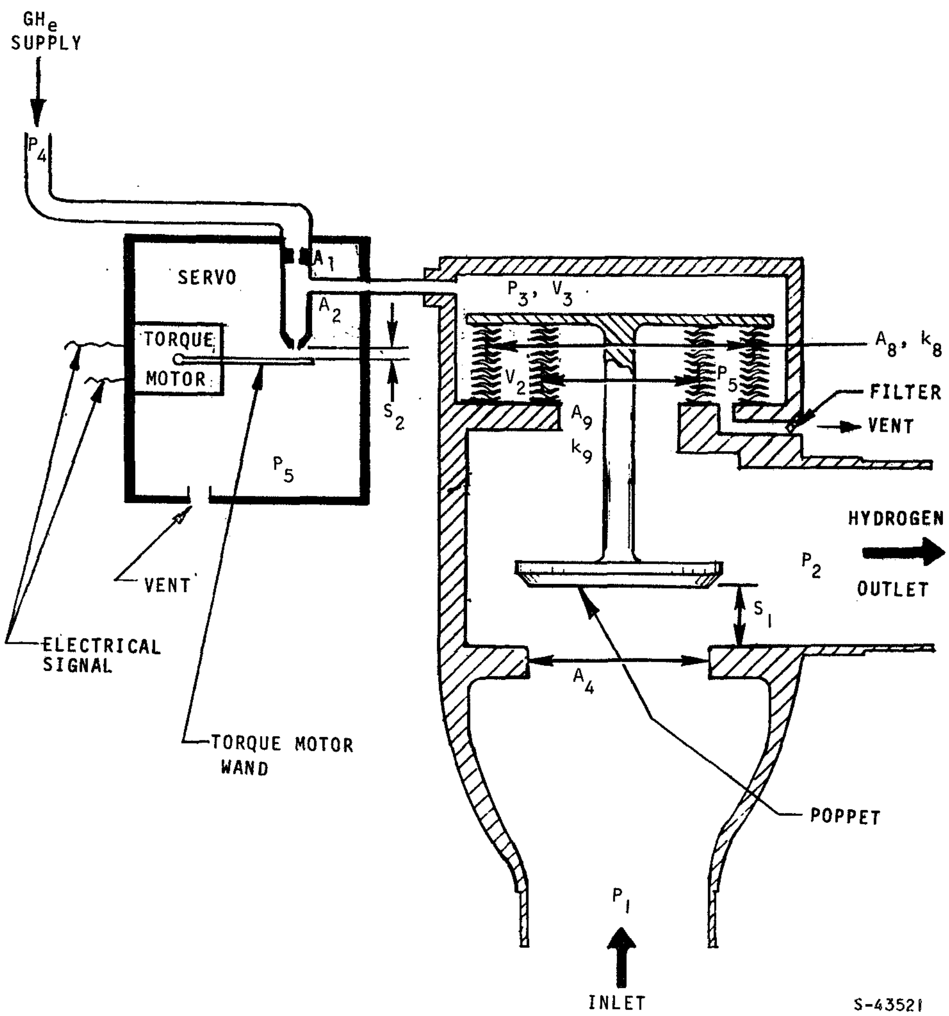


Figure A-4. Fuel Dump Valve Schematic (PN 393140)

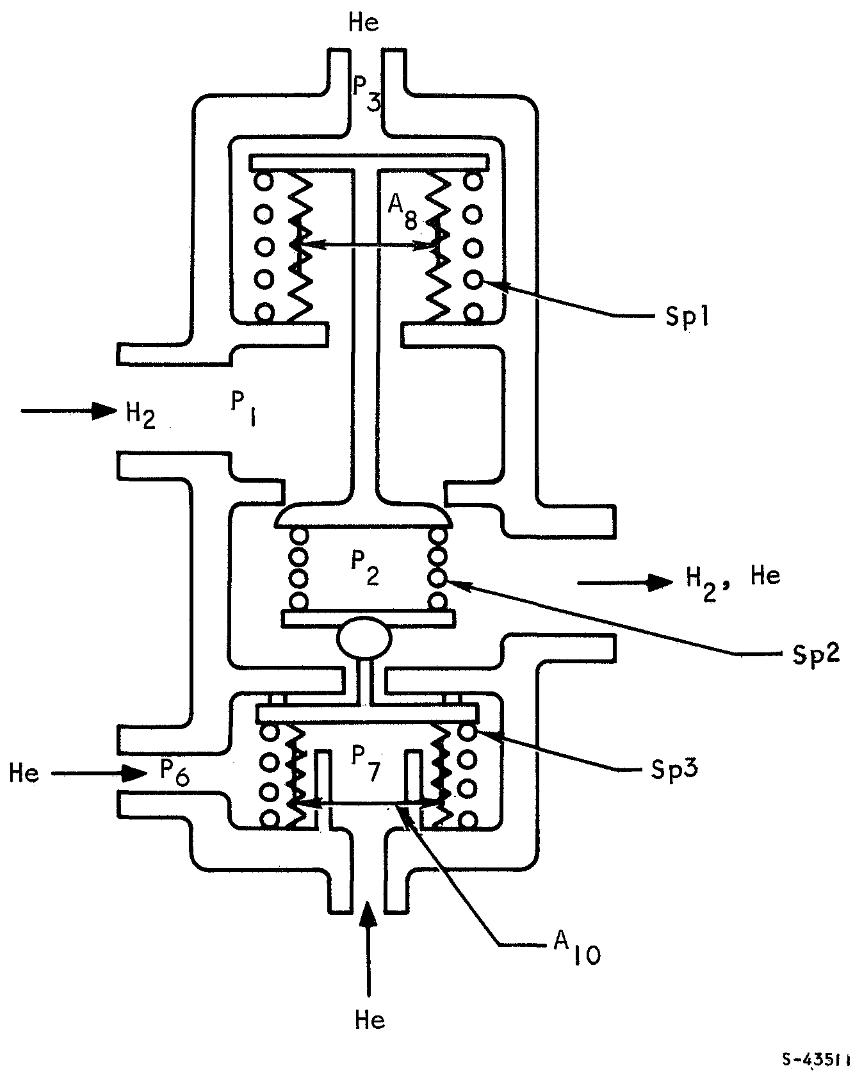


Figure A-5. Purge and Shutoff Valve Schematic (PN 395626)

Flow Area, PN 395626

The purge and shutoff valve area was determined as follows.

The assumption was made that, during operation, the hydrogen will be in both the liquid and the gaseous state.

Considering the hydrogen as a gas, the flow area was determined by the equation

$$A_{4_{(gas)_1}} = \frac{5.38\dot{\omega}\sqrt{T_1}}{C_d P_1 \sqrt{M}\left(\dfrac{\dot{\omega}\sqrt{T_1}}{P_1 A_4}\right)_{air,\ r_p}} \qquad \text{(A-2)}$$

where $r_p = \dfrac{P_2}{P_1} = 0.83$

$\left(\dfrac{\dot{\omega}\sqrt{T_1}}{P_1 A_4}\right)_{air,\ r_p} = 0.496$ for $\gamma = 1.4$

$M = 2$ (molecular weight of hydrogen)

$\dot{\omega} = 3.0$ lb/sec

$T_1 = 40^{o}R$

$C_d = 0.6$

$P_1 = 60$ psia

Then, $$A_{4_{(gas)_1}} = \frac{5.38(3.0)\sqrt{40}}{0.6(60)\sqrt{2}\ (0.496)} = 4.03 \text{ sq in.}$$

Considering the hydrogen as a liquid, the flow area was determined by the equation

$$A_{4_{(liq)_1}} = \frac{1.495\dot{\omega}}{C_d\sqrt{(\Delta P)\rho}} \qquad \text{(A-3)}$$

where $\dot{\omega} = 3.0$ lb/sec

$C_d = 0.6$

$\rho = 4.42$ lb/cu-ft

$\Delta P = 10$ psi

Therefore,

$$A_{4(liq)_1} = \frac{1.495(3.0)}{0.6\sqrt{10(4.43)}} = 1.125 \text{ sq in.}$$

Assuming a shaft diameter of 0.20 in., the area of the shaft is 0.03 sq in. This area is then added to the areas previously determined for the gaseous and liquid states. The required seat areas are therefore

$$A_{4_2} = A_{4_1} + A_{SHFT} \tag{A-4}$$

$$A_{4(gas)_2} = 4.03 + 0.03 = 4.06 \text{ sq in.}$$

$$A_{4(liq)_2} = 1.125 + 0.03 = 1.155 \text{ sq in.}$$

The seat diameters are

$$d_{(gas)} = \sqrt{\frac{4}{\pi}(4.06)} = 2.27 \text{ in.}$$

$$d_{(liq)} = \sqrt{\frac{4}{\pi}(1.155)} = 1.47 \text{ in.}$$

To permit adequate flow through the valve without overdesigning, an average seat diameter of the two extremes was selected. The design seat diameter is then 1.88 in., which provides a flow area of 2.75 sq in.

The helium flow area of the purge and shutoff valve was determined by using Equation (4-2).

$$A_4 = \frac{5.38\dot{\omega}\sqrt{T_1}}{C_d P_1 \sqrt{M}\left(\frac{\dot{\omega}\sqrt{T_1}}{P_1 A_4}\right)_{air,\ r_p}}$$

where $r_p = \frac{60}{575}$ = choked

$\left(\frac{\dot{\omega}\sqrt{T_1}}{P_1 A_4}\right)_{air,\ r_p}$ = 0.564 for γ = 1.67

M = 4 (molecular weight of helium)

$\dot{\omega}$ = 0.058 lb/sec

T_1 = 260°R

P_1 = 575 psia

C_d = 0.6

then

$$A_4 = \frac{5.38(0.058)\sqrt{260}}{0.6(575)\sqrt{4}\ (0.564)} = 0.0131 \text{ sq in.}$$

Assume a poppet shaft diameter of 0.05 in. The shaft area is then 0.002 sq in. The poppet seat area is

$$A_4 = 0.0131 + 0.002 = 0.0151 \text{ sq in.}$$

and the seat diameter is

$$d = \sqrt{\frac{4}{\pi}(0.0151)} = 0.139 \text{ in.}$$

POPPET STROKE

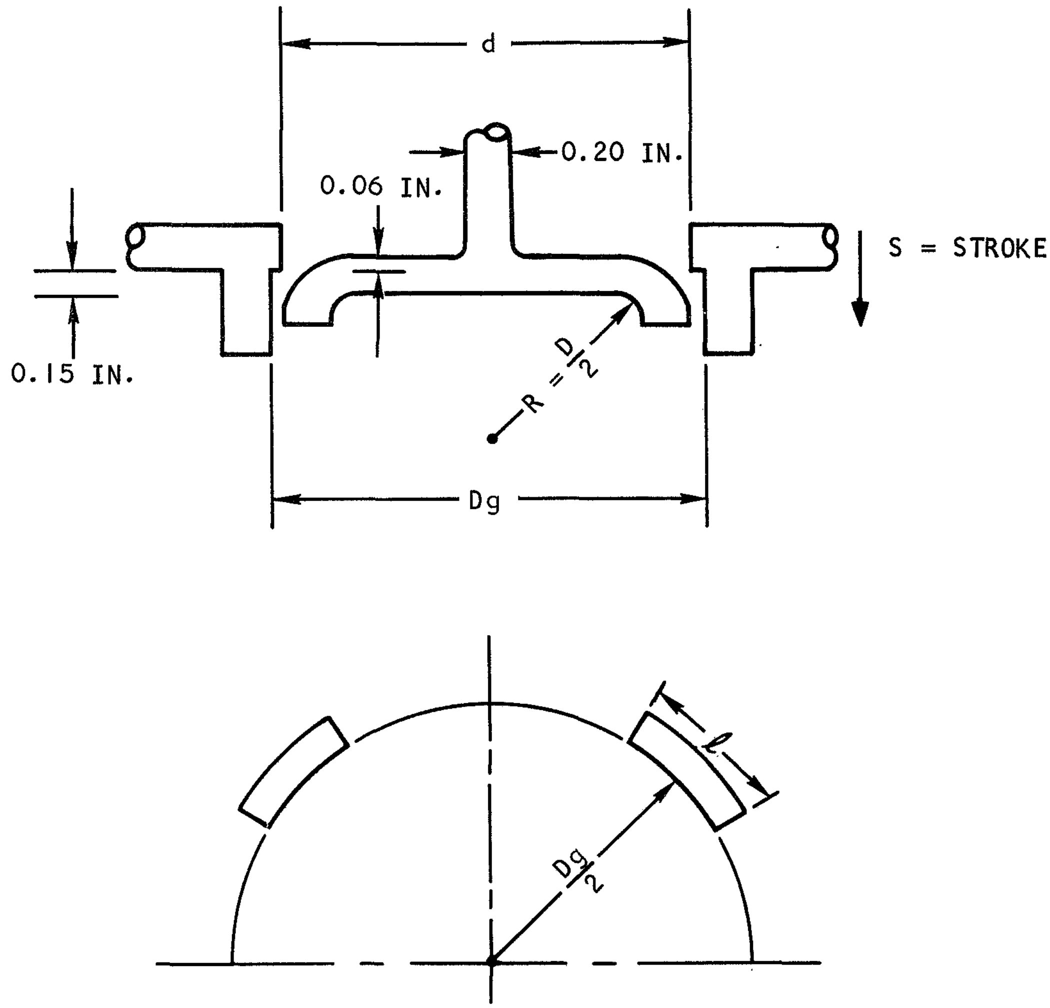
The minimum stroke required of a flat poppet modulating valve is determined from the following relationship:

$$S = \frac{A_4}{\pi d} \qquad (A-5)$$

Parameter \ Valve PN	393088	393090	393094	393140
A_4, sq in.	1.539	2.46	0.568	4.08
d, in.	1.40	1.77	0.850	2.28
S_{min}, in.	0.35	0.443	0.2125	0.57
S_{design}, in.	0.50	0.50	0.35	0.60

Figure A-6 shows the flow area vs stroke relationship for the valves noted.

The purge and shutoff valve (PN 395626), which is not a modulating valve, has its poppet design criteria determined as follows.



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A d/D ratio of approximately 0.7 was assumed to provide a valve gain which is nearly constant with stroke (refer to Chapter 3, Reference A-1)*. For $d = 1.88$ in., the ball diameter, D, is 2.67 in. From the geometry shown above the guide diameter, Dg, is 2.202 in.

The poppet stroke is determined from

$$A_4 = \pi d(S - 0.06) \tag{A-6}$$

*Reference A-1. Anderson, Blaine W., "The Analysis and Design of Pneumatic Systems," Wiley, 1967.

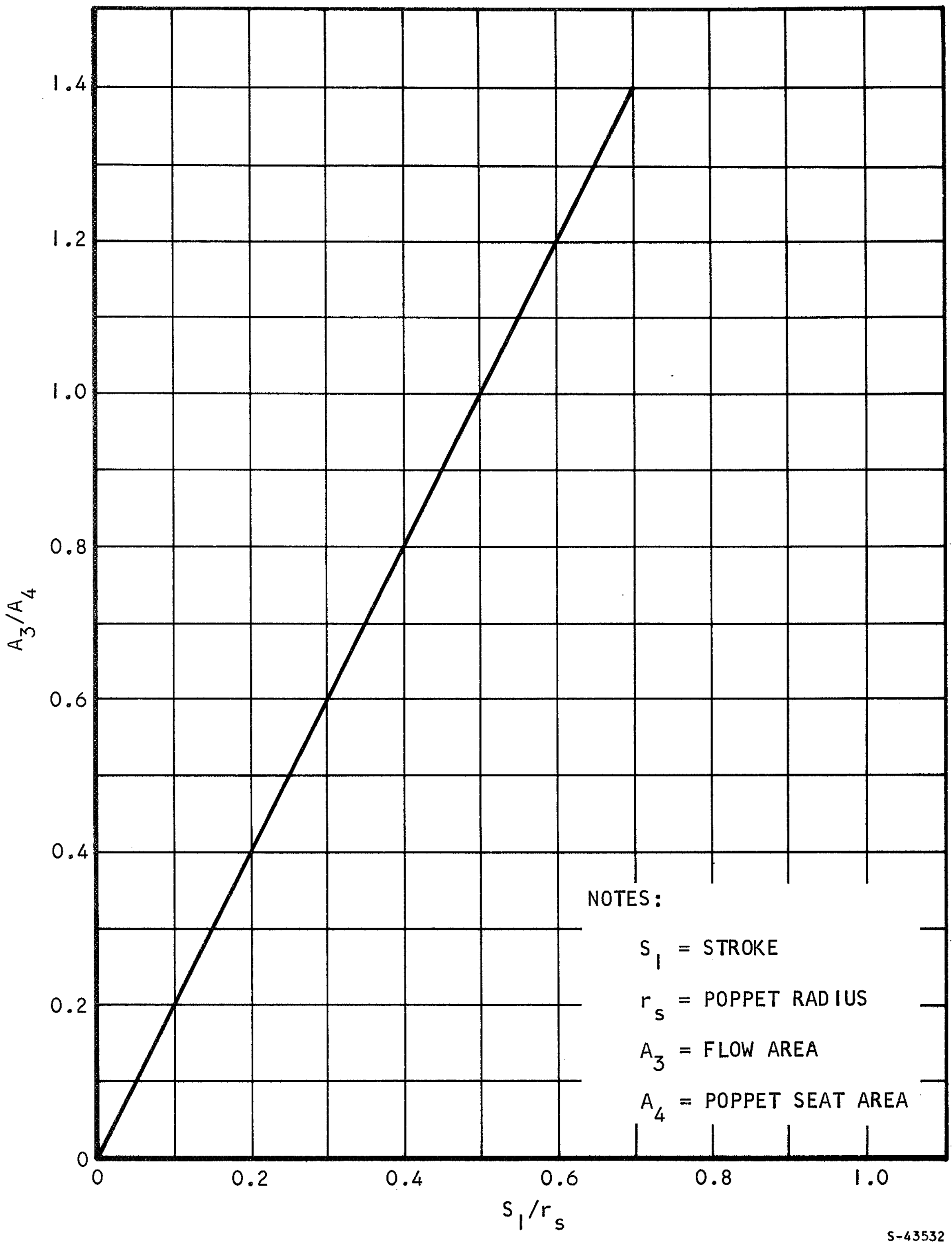


Figure A-6. Flow Area vs Stroke for Flat Poppets

Rearranging Equation (A-6) gives

$$S = \frac{A_4}{\pi d} + 0.06 \tag{A-7}$$

Therefore,

$$S = \frac{2.75}{\pi(1.88)} + 0.06 = 0.498 \text{ in.}$$

The maximum design stroke is then 0.5 in.

The flow area between the guides is found from

$$A_{11} = \pi Dg(S + 0.15) - 4\ell(S + 0.15) \tag{A-8}$$

where ℓ is assumed to be 0.45 in.

Then,

$$A_{11} = \pi(2.202)(0.5 + 0.15) - 4(0.45)(0.5 + 0.15)$$

$$= 3.34 \text{ sq in.}$$

POPPET AND ACTUATION FORCES

For the modulating valves, which include all but the purge and shutoff valve (PN 395626), the poppet forces were determined for several positions of poppet stroke. Following this, the actuator was sized to provide the required actuation forces.

Poppet Forces (PN's 393090, 393088, 393094)

Sample calculations for a 0.25 in. poppet stroke (PN 393090) are given below. Values used in the calculation are as follows:

$$r_s = 0.94 \text{ in.}$$

$$A_4 = 2.77 \text{ sq in. (max. flow area)}$$

$$A_{12} = 1.69 \text{ sq in.}$$

$$S_1/r_s = \frac{0.25}{0.94} = 0.266$$

$$A_3/A_4 = 0.53 \text{ (value from Figure A-6)}$$

$$A_3 = (0.53)(2.77) = 1.465$$

$$A_3/A_{12} = \frac{1.465}{1.69} = 0.867$$

$P_2/P_1 = 0.752$ (value derived from Figure A-7--$P_2/P_1 \equiv P_y/P_z$)

$P_1 - P_2 = \Delta P$ at 500 psia inlet pressure $= 500 - (0.752)(500)$

$= 500 - 376 = 124$ psi

$= \Delta P$ at 550 psia inlet pressure $= 550 - (0.752)(550)$

$= 550 - 414 = 136$ psi

At a 500 psia inlet pressure,

$$\frac{F_p}{(P_1 - P_2)(A_4)} = 1 - \frac{A_3}{A_4} = (1 - 0.53) \tag{A-9}$$

$= 0.47$ (equation based on empirical test data)

and $$F_p = (0.47)(2.77)(124) = 161.5 \text{ lb}$$

At a 550 psia inlet:

$$F_p = (0.47)(2.77)(136) = 177 \text{ lb}$$

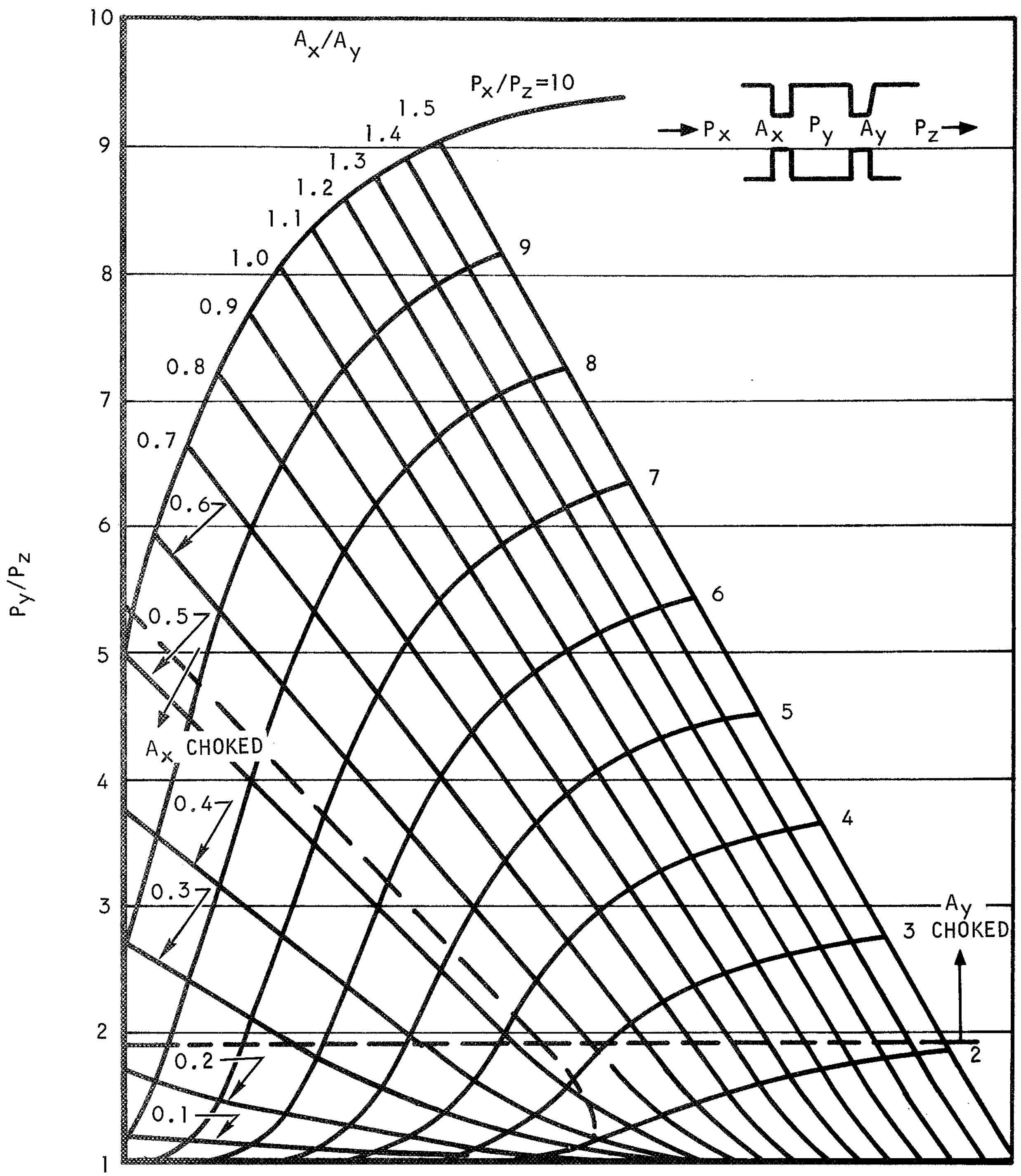
The ratio of the poppet force at the selected point to the maximum total poppet force is

$$\frac{F_T}{F_{T(\max.)}} = \frac{F_P}{(P_1 - P_2)(A_4)} \cdot \frac{(P_1 - P_2)}{P_1} = \frac{(0.47)(124)}{500} = 0.1163 \tag{A-10}$$

The ratio of the flow at the selected point to the maximum valve flow capacity is

$$\frac{\dot{\omega}}{\dot{\omega}_{\max.}} = \frac{1.36}{1.66} = 0.821$$

Results of the preceding calculations and for PN's 393088 and 393094 are presented in Figures A-8 through A-20.



PRESSURE DIVIDER CHARACTERISTICS (k= 1.4)

A-31884

Figure A-7. Pressure Divider Characteristics (k = 1.4)

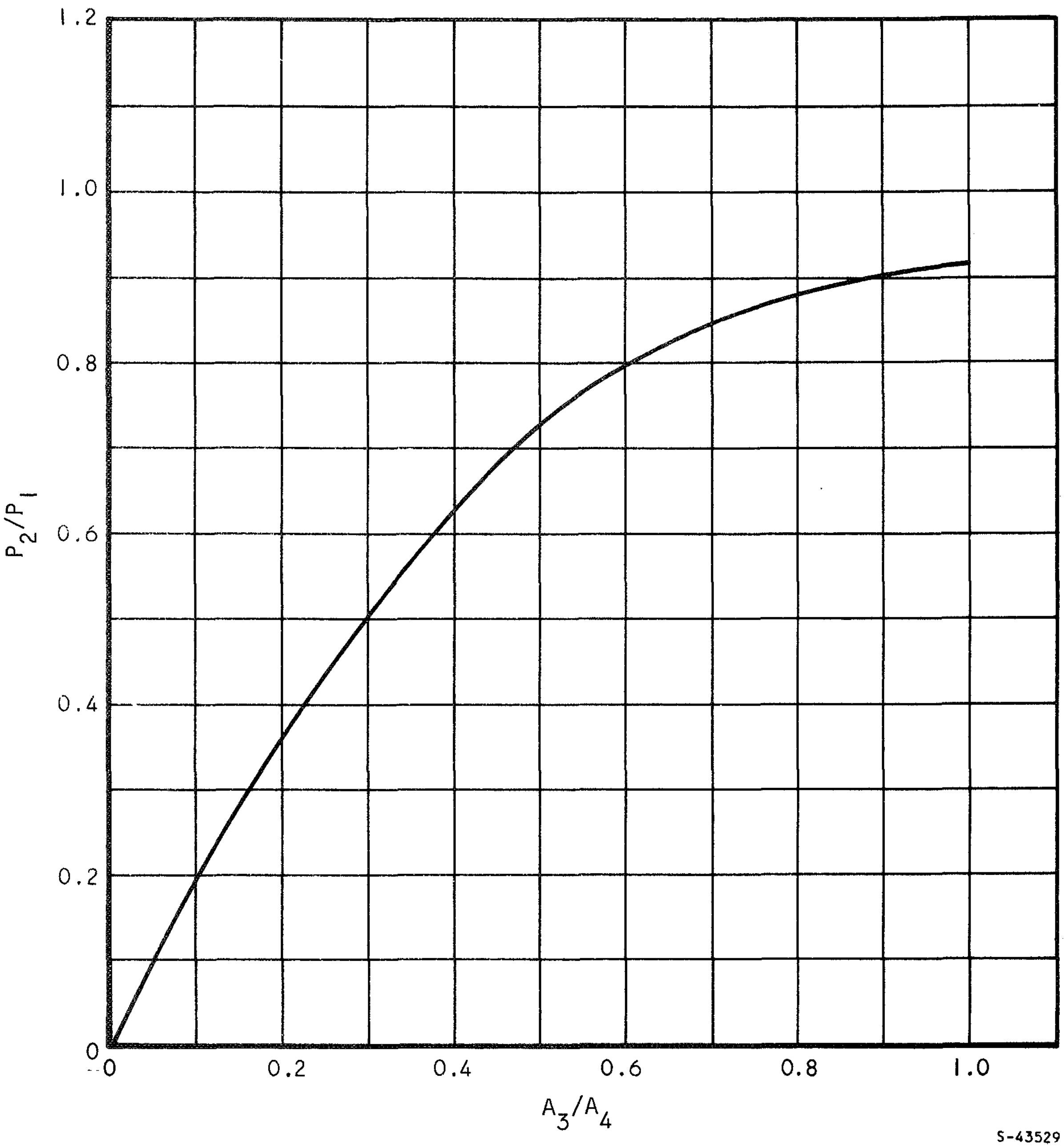


Figure A-8. Pressure Ratio vs Flow Area (PN 393090)

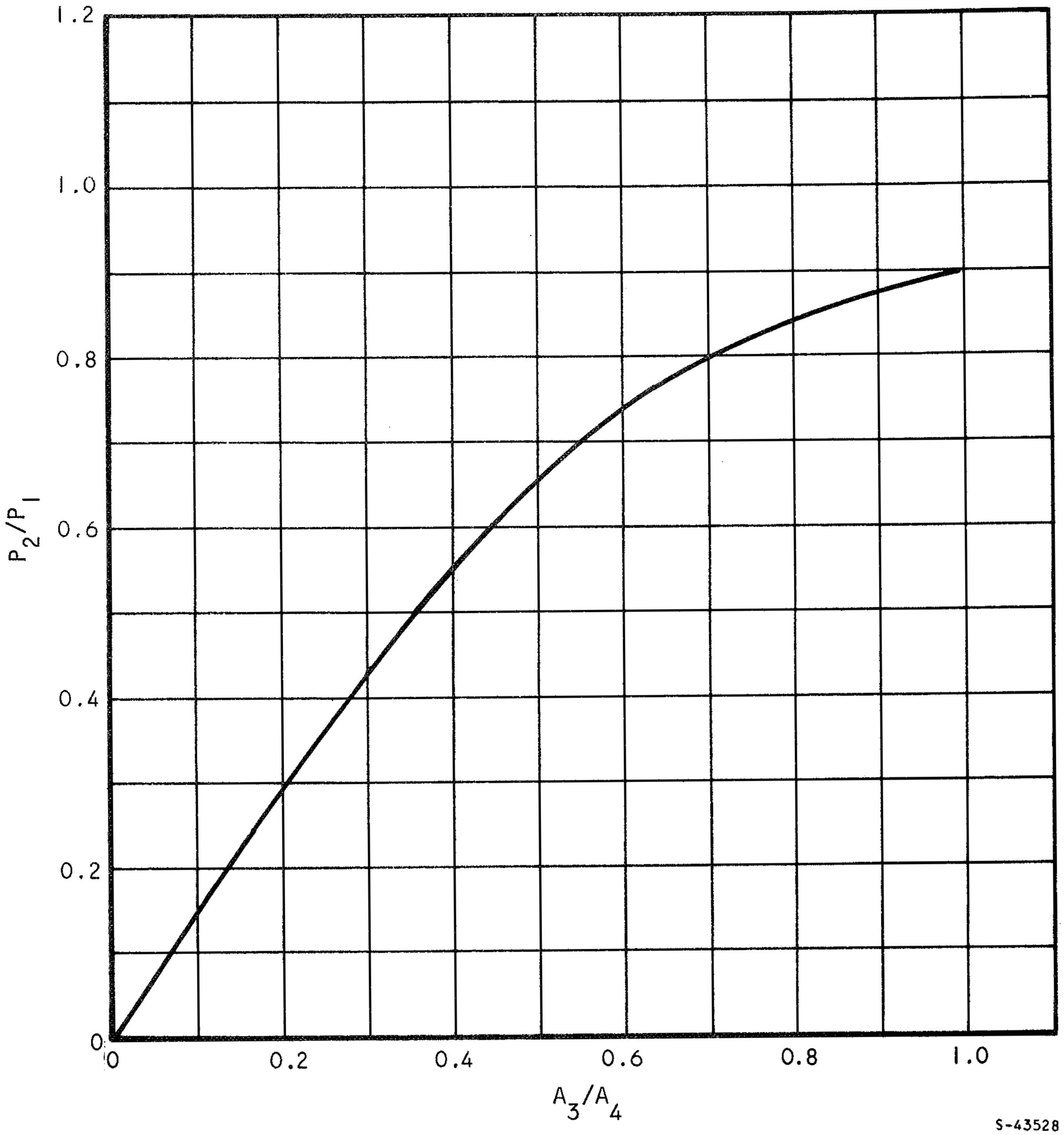


Figure A-9. Pressure Ratio vs Flow Area (PN 393088)

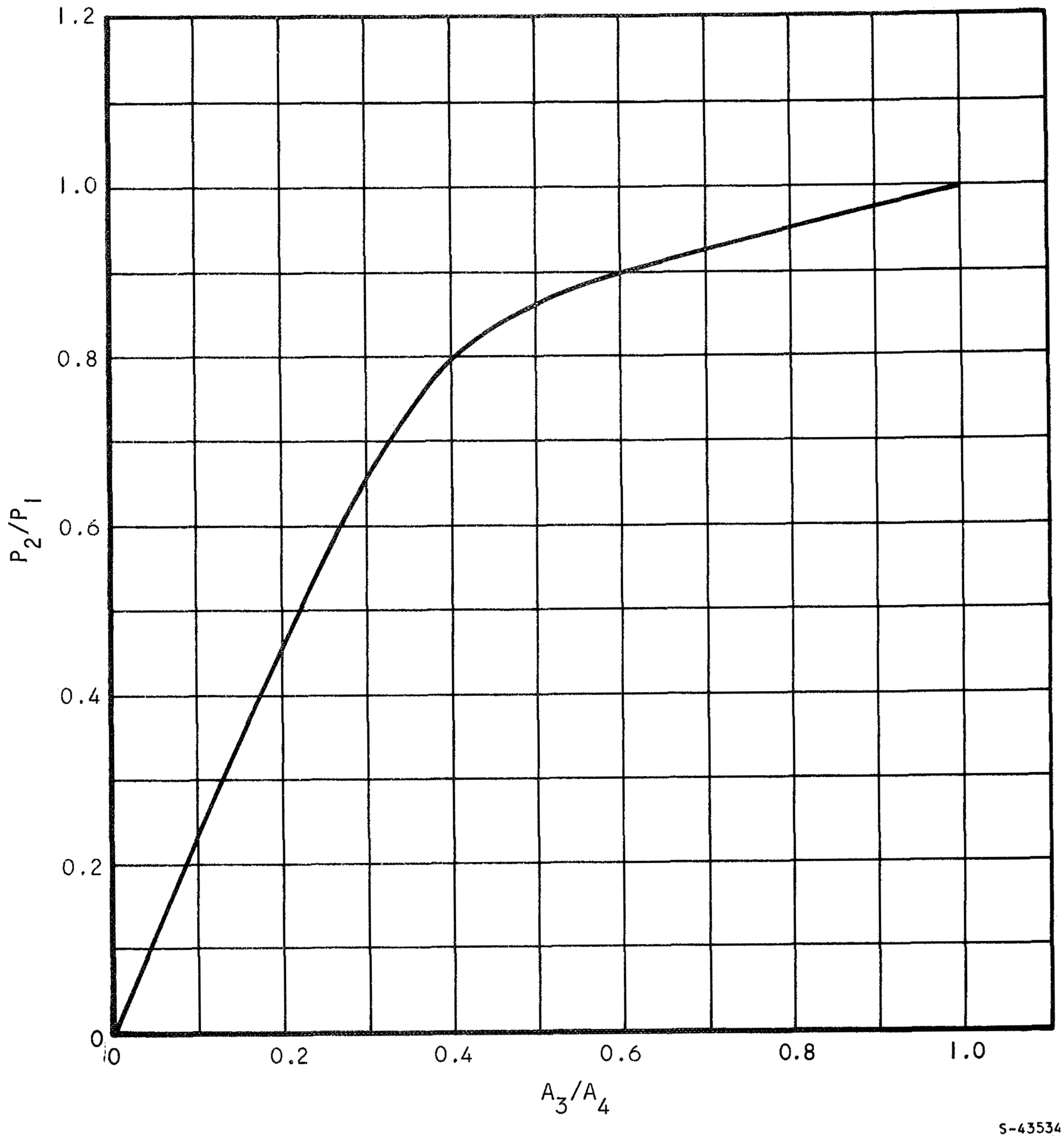


Figure A-10. Pressure Ratio vs Flow Area (PN 393094)

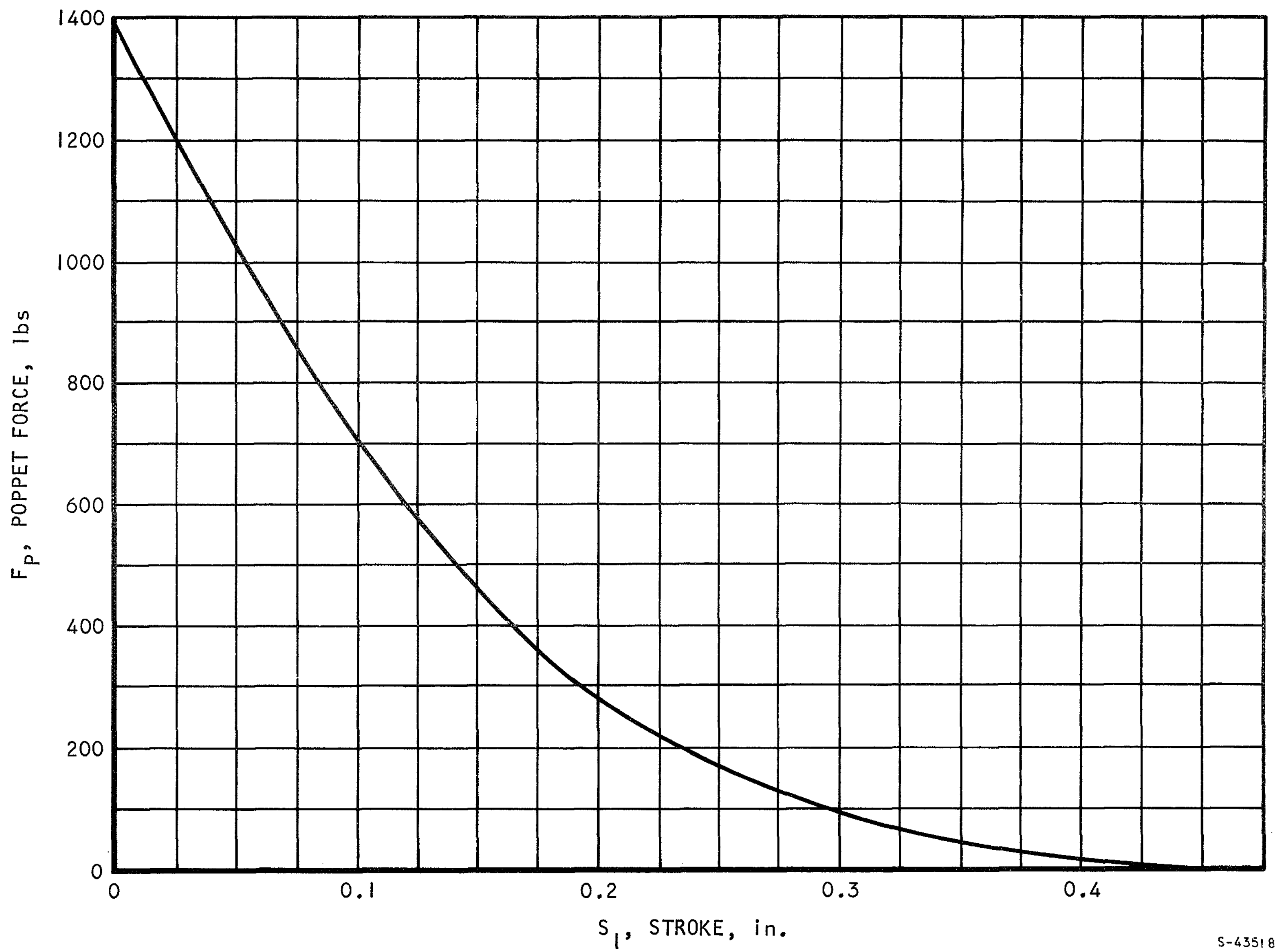


Figure A-11. Poppet Force vs Stroke (PN 393090)

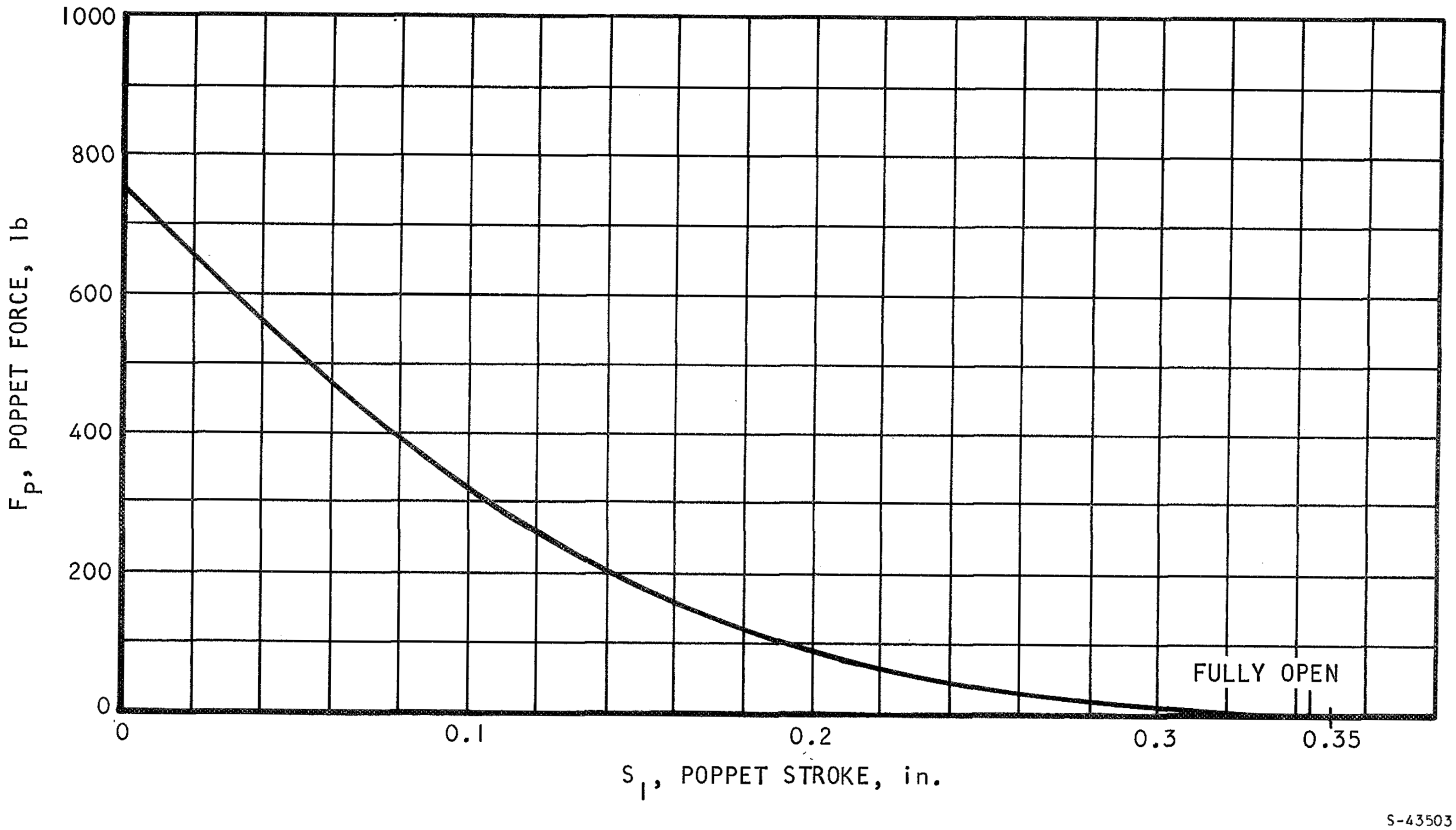


Figure A-12. Poppet Force vs Stroke (PN 393088)

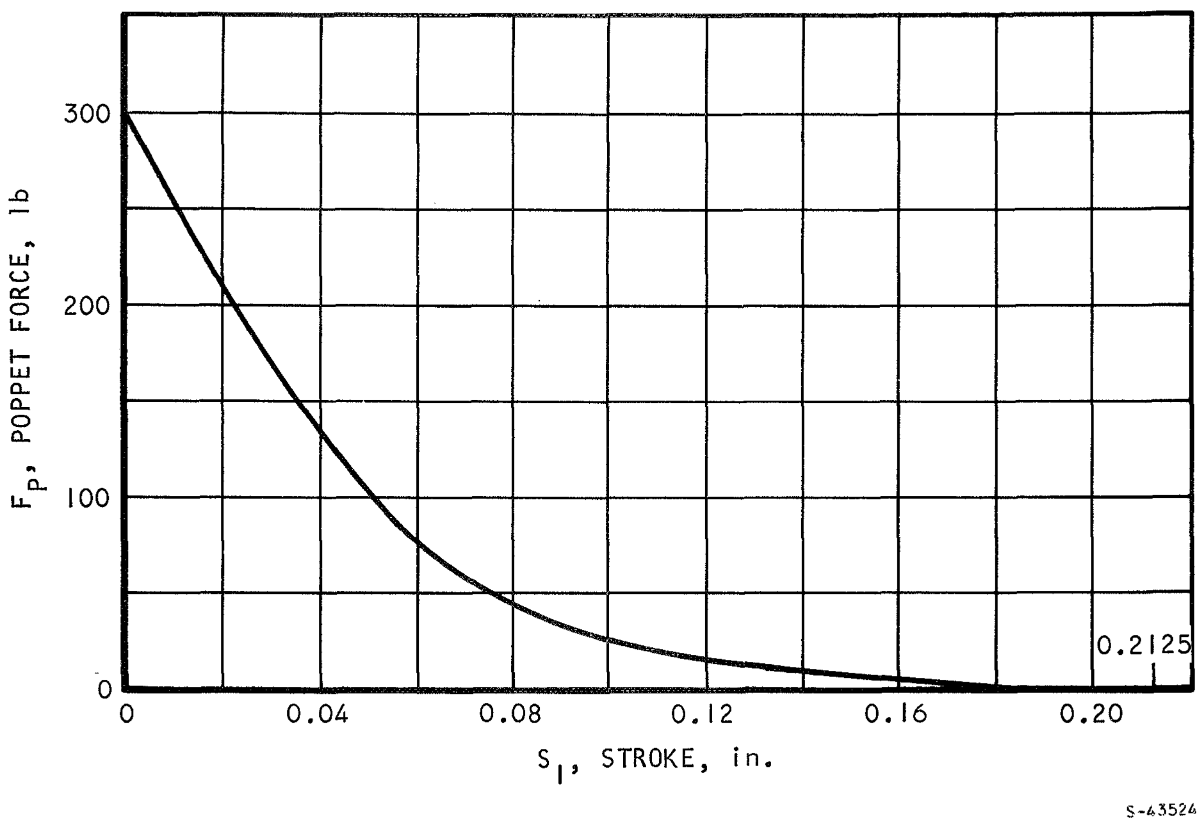


Figure A-13. Poppet Force vs Stroke (PN 393094)

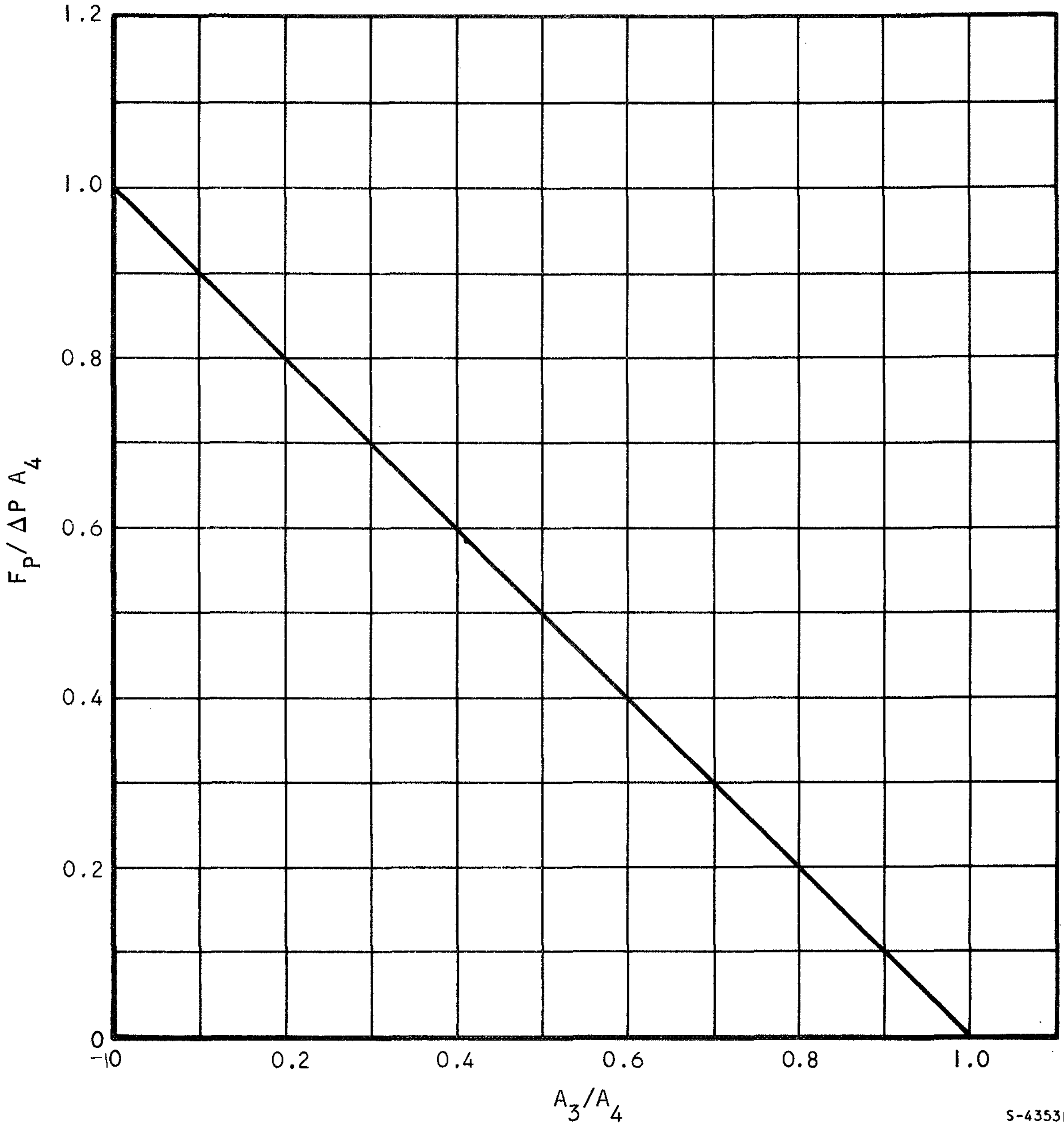


Figure A-14. Actuator vs Flow Area (PN's 393088, 393090, 393094)

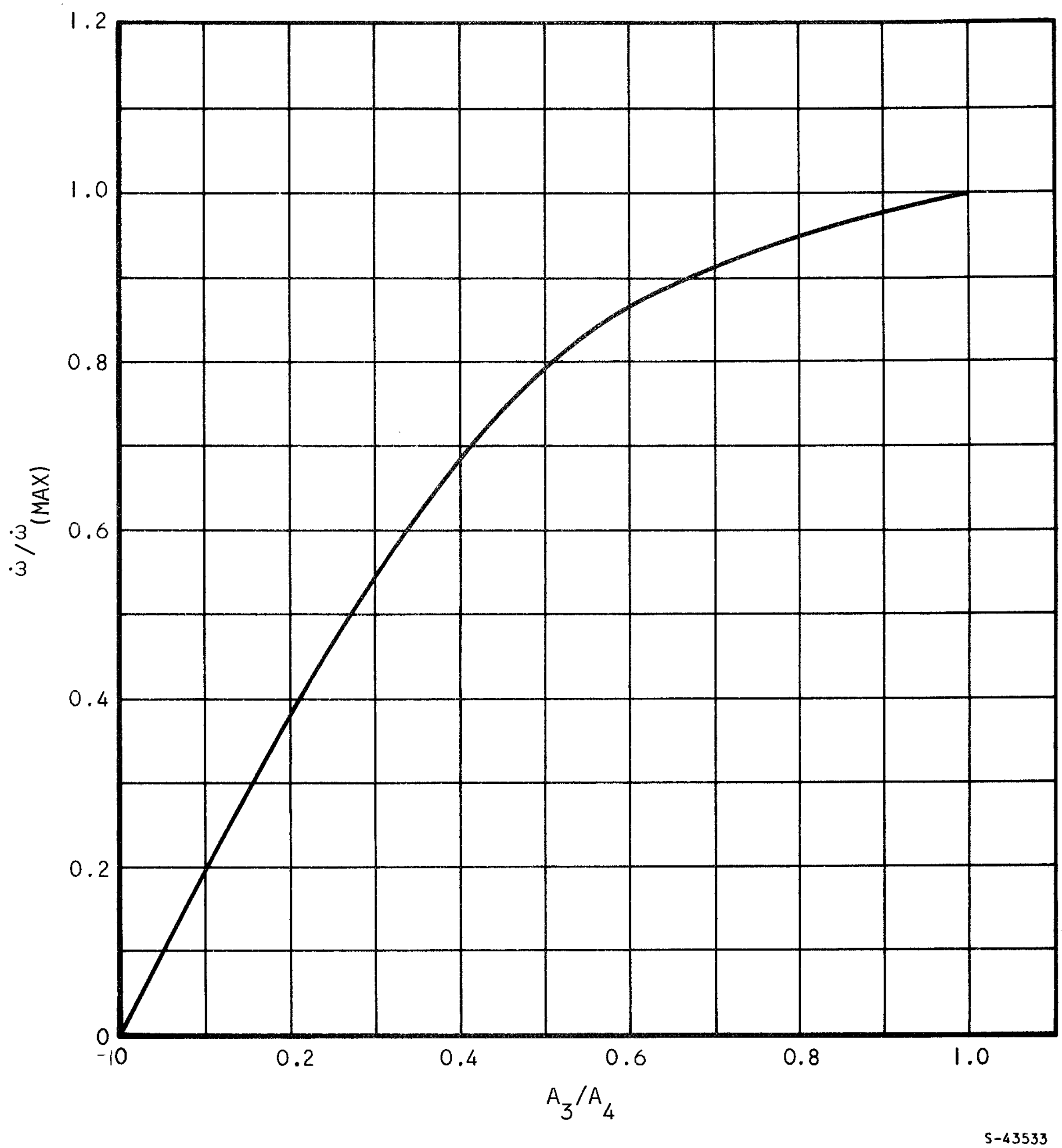


Figure A-15. Flow Ratio vs Flow Area (PN 393090)

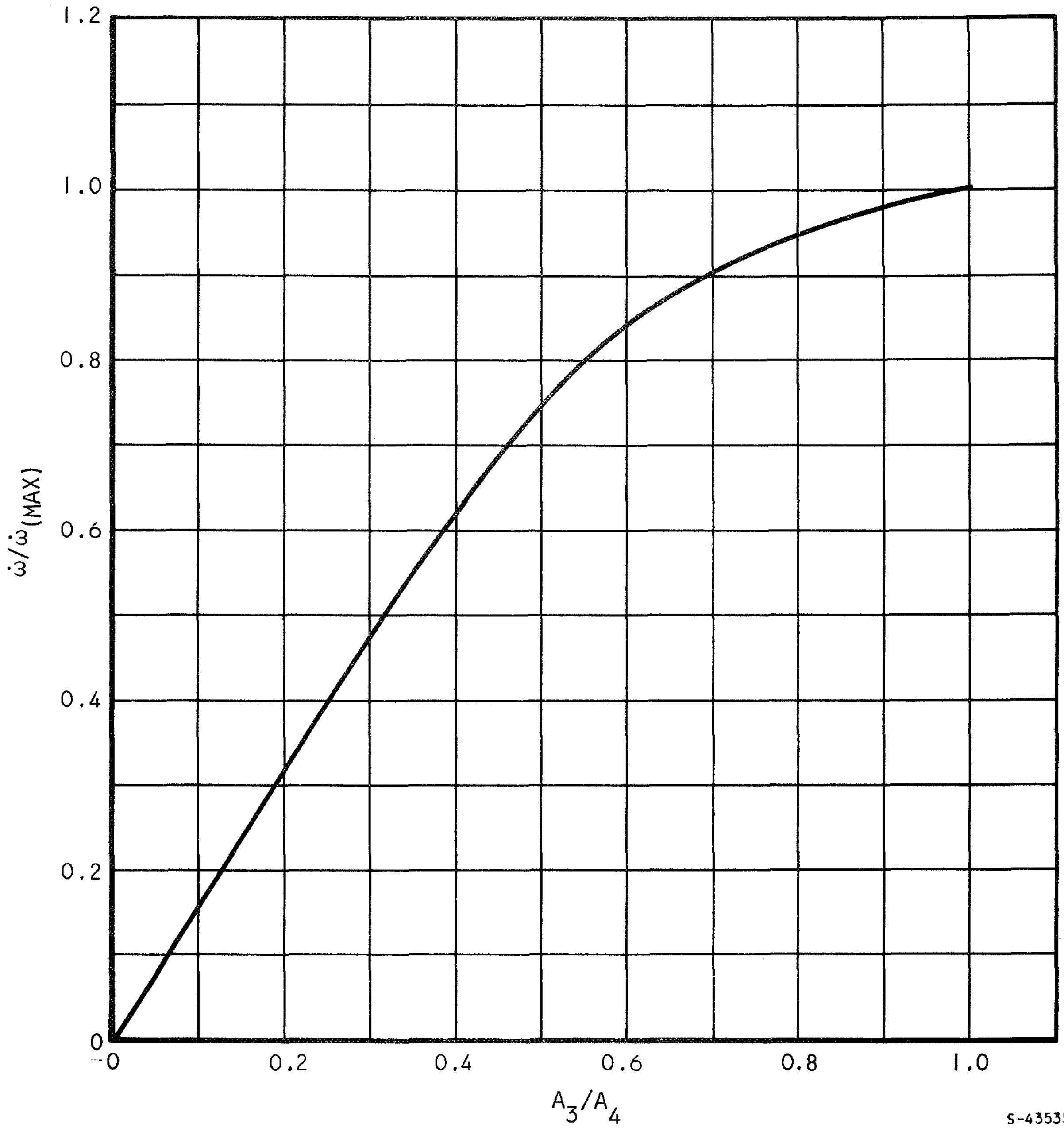


Figure A-16. Flow Ratio vs Flow Area (PN 393088)

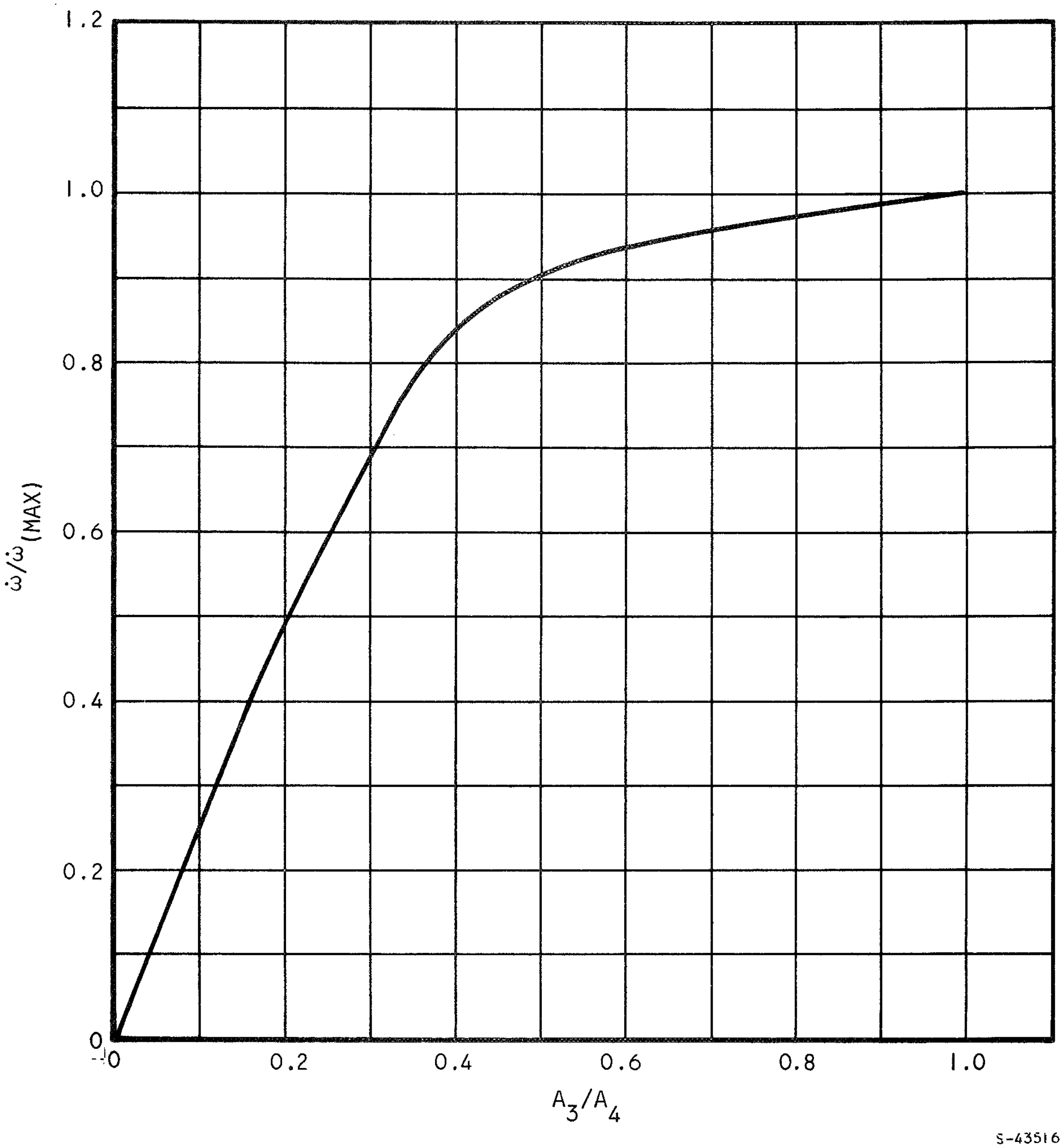


Figure A-17. Flow Ratio vs Flow Area (PN 393094)

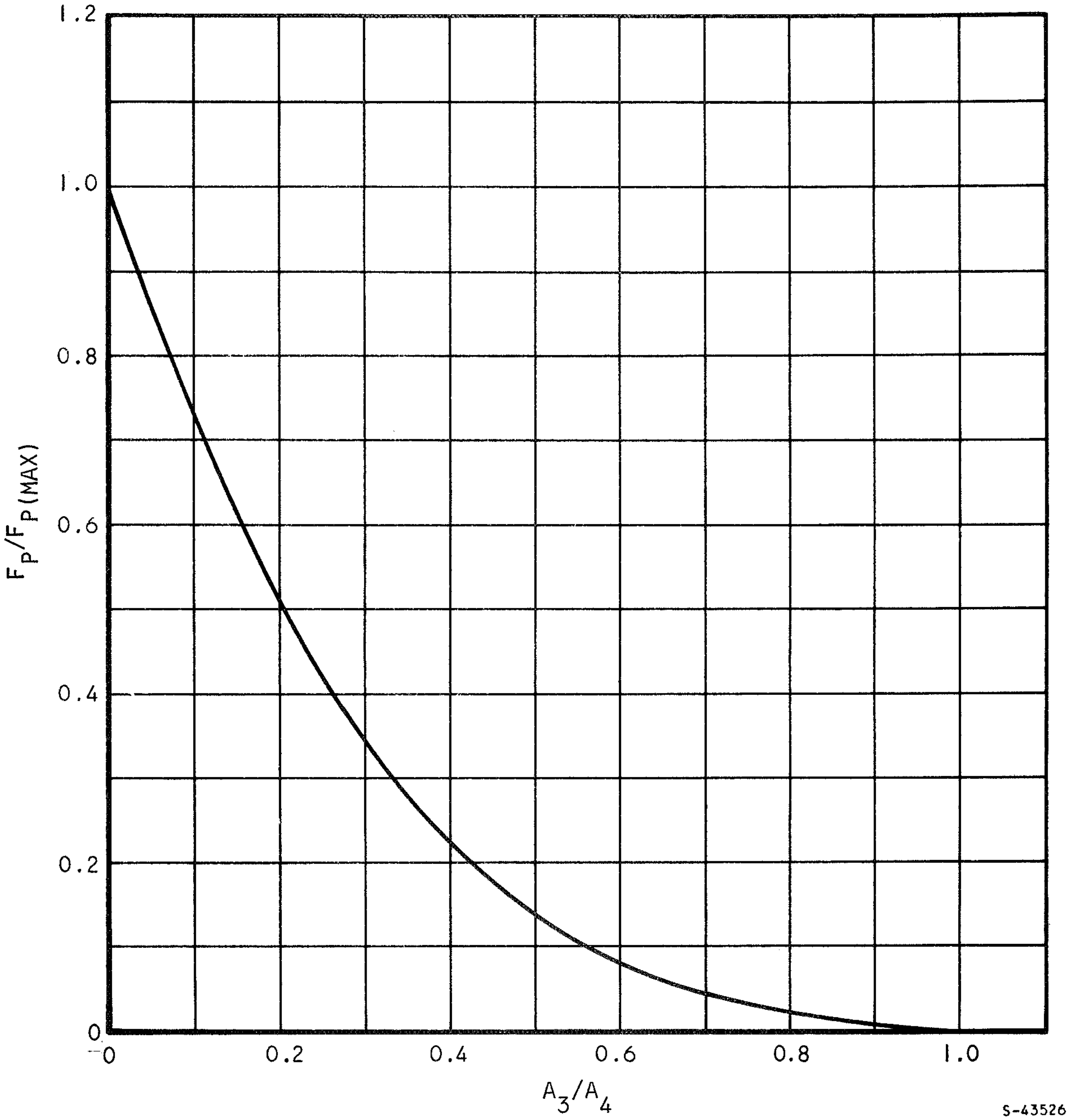


Figure A-18. Force Ratio vs Flow Area (PN 393090)

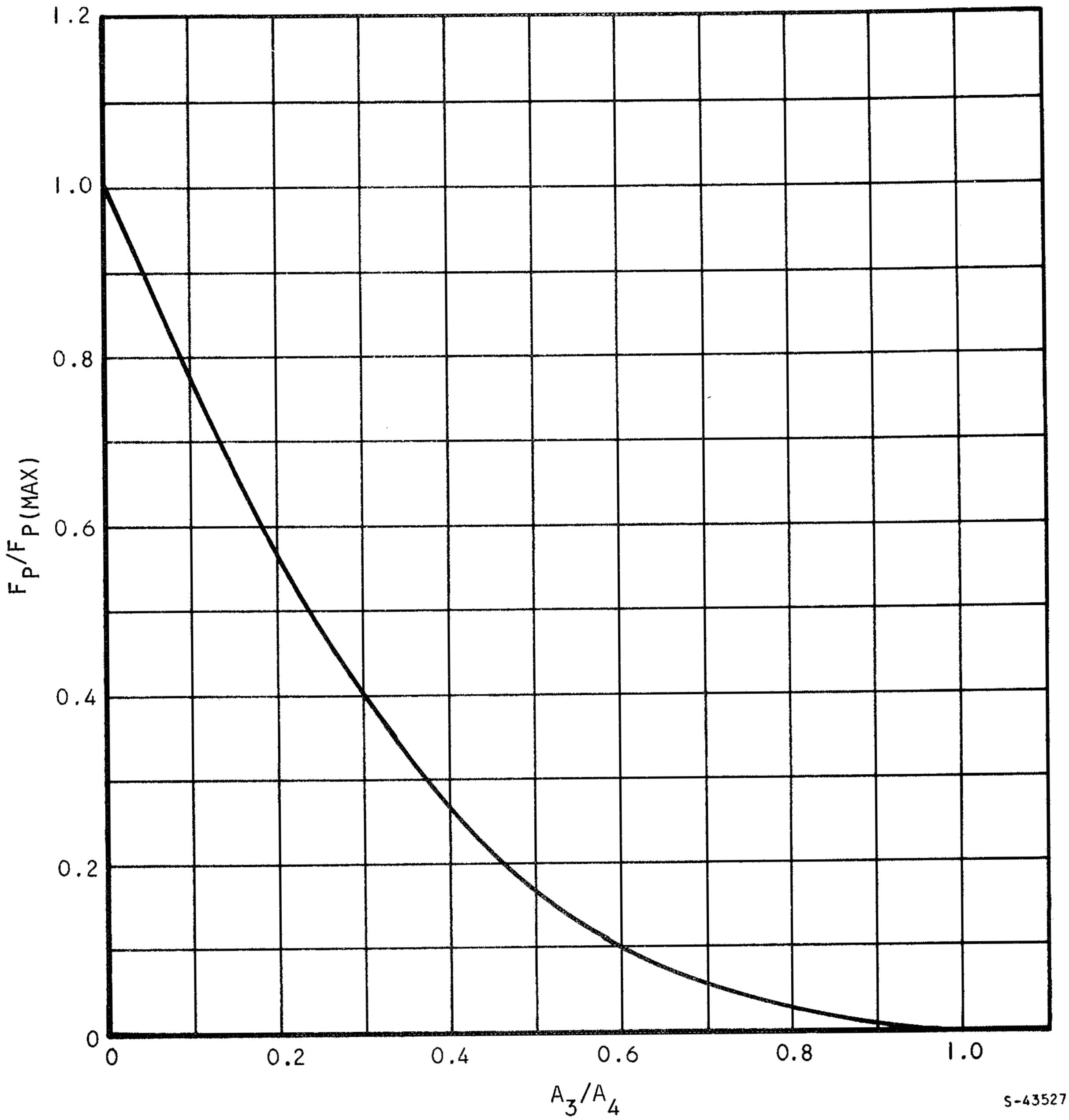


Figure A-19. Force Ratio vs Flow Area (PN 393088)

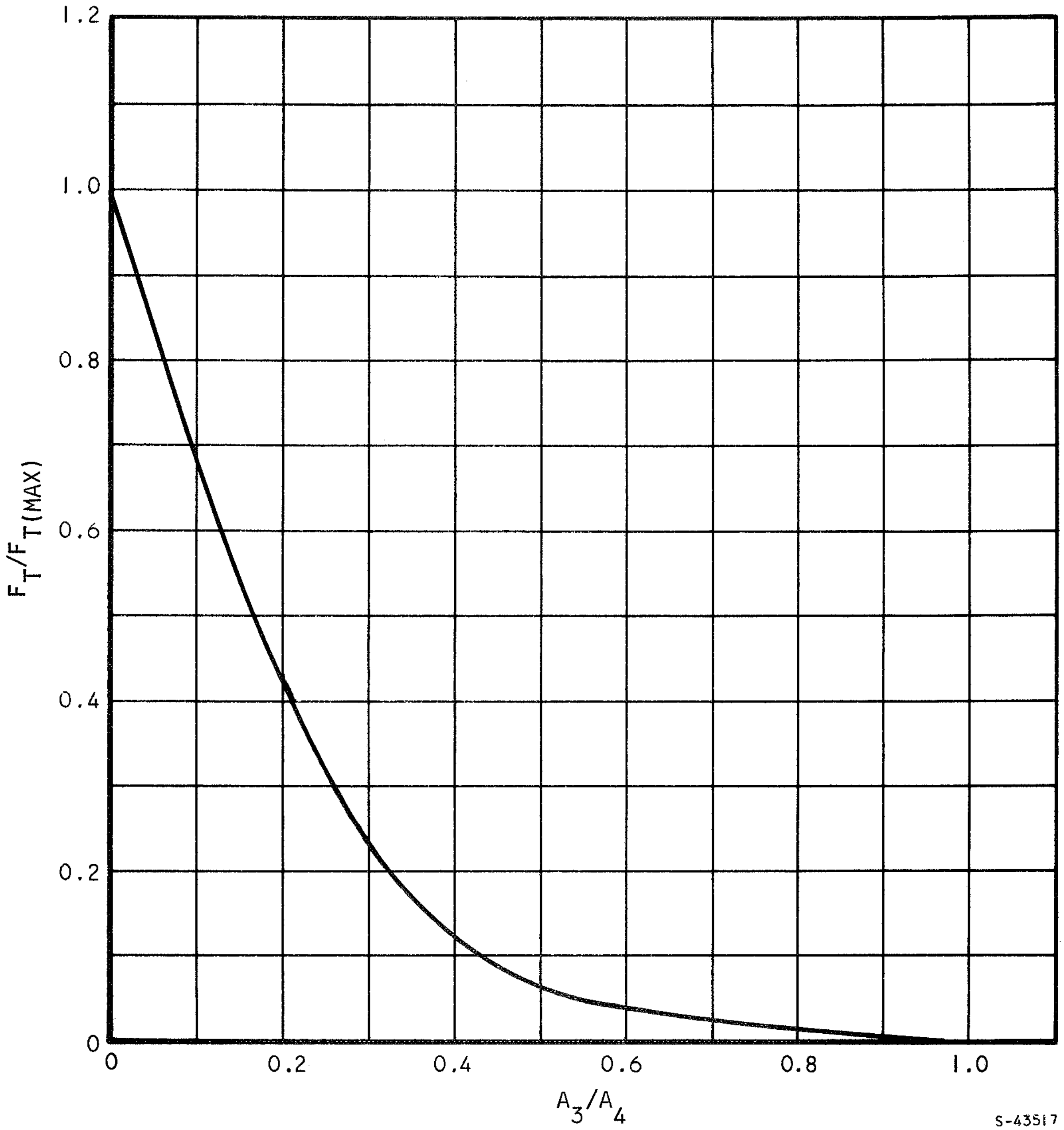


Figure A-20. Force Ratio vs Flow Area (PN 393094)

Actuation Forces (PN's 393090, 393088, 393094)

After determining the poppet forces required, a bellows area sufficient to provide the required actuation forces utilizing the available actuation gas supply pressures was selected from available sizes. The bellows areas selected are tabulated below.

PN	Bellows Area, sq in.	Bellows Spring Rate, lb/in.	Preload, lb
393090	3.98	104	10.4
393088	1.92	75	7.5
393094	1.92	75	---

By utilizing the curves from the previous section, and by including the bellows spring rate forces and the preloads, the total required actuation forces were determined. These are plotted in Figures A-21 through A-23 as actuator force vs valve flowrate. For the turbine control valve, PN 393094, it is more convenient, however, to plot actuation ΔP vs flow ΔP.

Poppet and Actuation Forces (PN 393140)

Operation of the fuel dump valve, PN 393140, is somewhat different because of failure modes and flow requirements. Two bellows assemblies are utilized as shown in Figure A-4. The force balance equation for this configuration is as follows:

$$\text{(seating force)} + (P_1 - P_2)^* \left[A_4\left(1 - \frac{S_1}{0.57}\right)\right] + P_2A_9 + (k_8 + k_9)(0.57 - S_1) + P_5(A_8 - A_9) = P_3A_8 \qquad \text{(A-11)}$$

In determining the required bellows areas, the smaller area, A_9, was initially selected as 1.92 sq in. (same bellows as for the 393094 valve actuator) with a spring rate of 75 lb/in. Other parameters for Equation (4-11) for the maximum force (closed valve) position are:

$$\text{seating force} = \text{desired at } 20(\pi 2.28) = 143 \text{ lb}$$

$$A_4 = 4.08 \text{ sq in.}$$

*Assumed poppet force area based upon test data

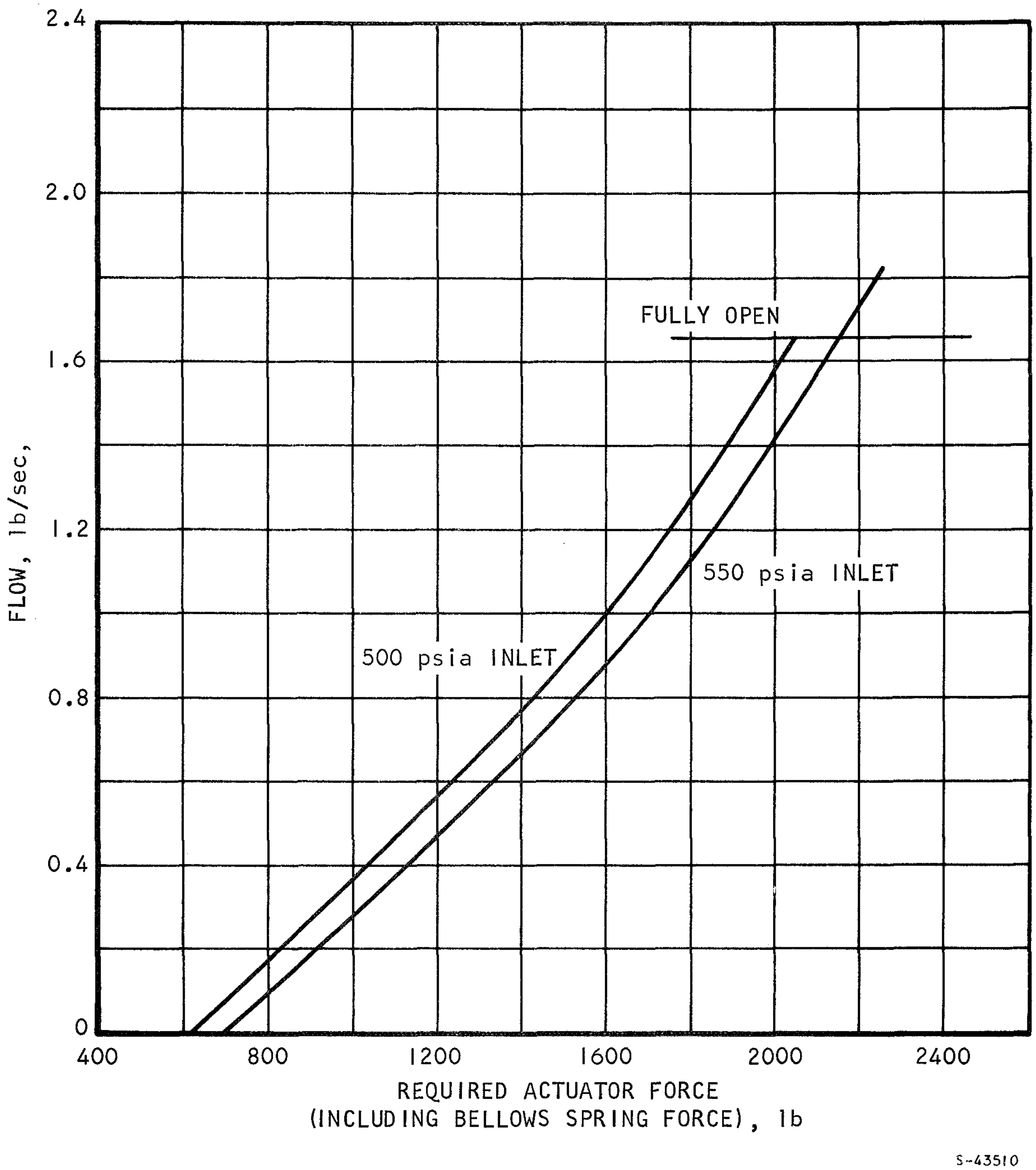


Figure A-21. Flow vs Required Actuator Force (PN 393090)

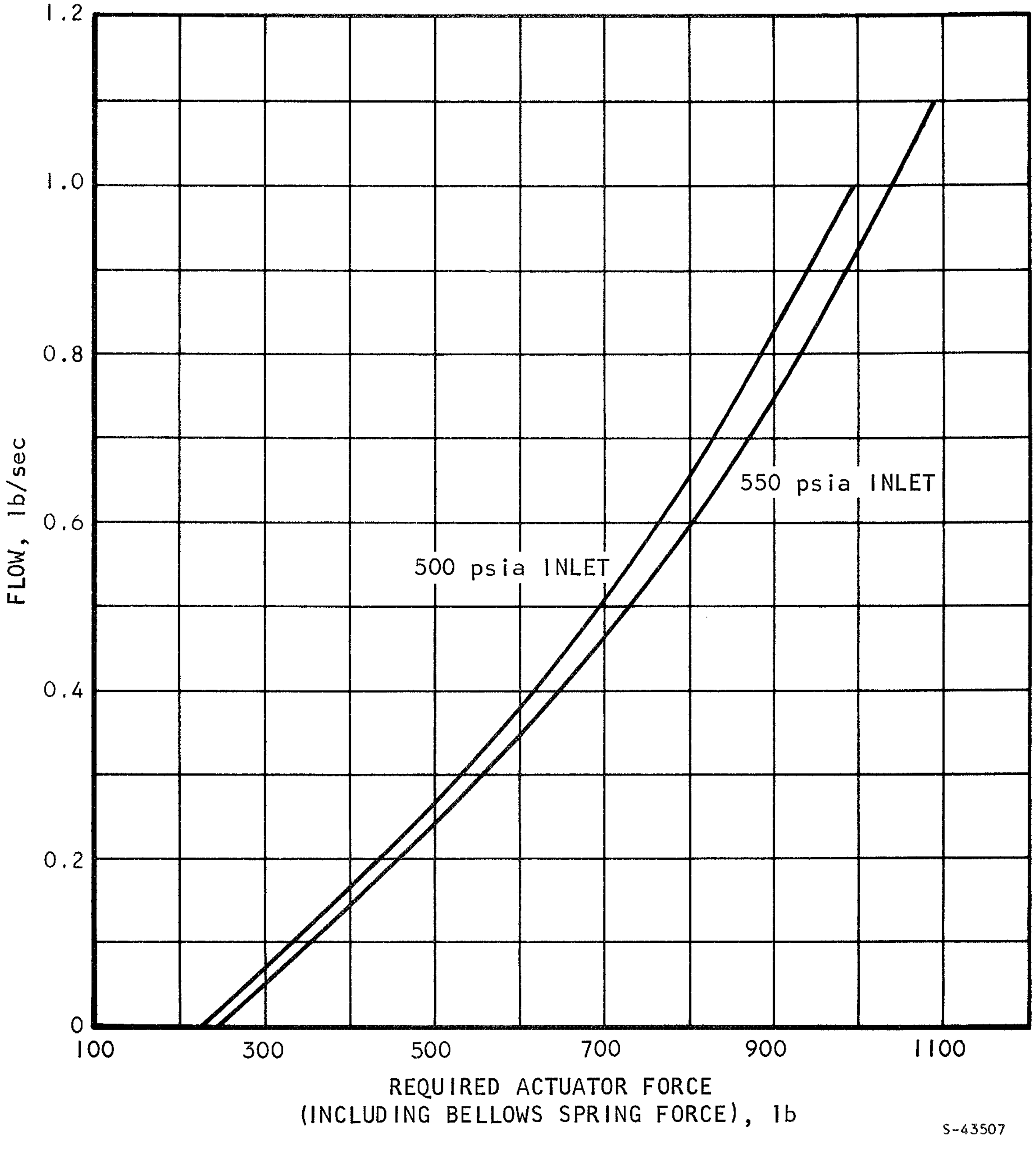


Figure A-22. Flow vs Required Actuator Force (PN 393088)

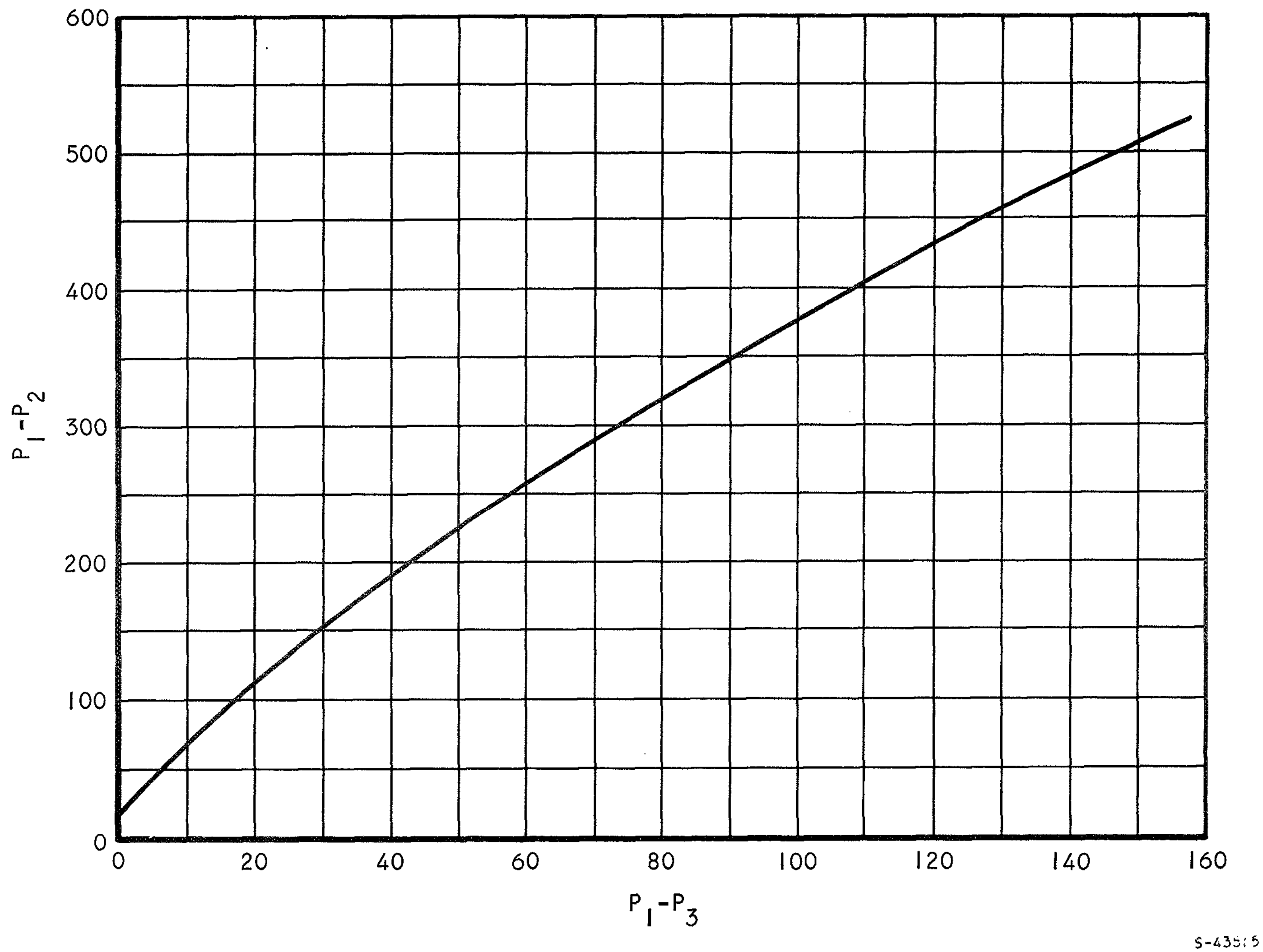


Figure A-23. Flow ΔP vs Actuator ΔP (PN 393094)

$$P_1 = 550 \text{ psia}$$

$$P_2 = P_5 = S_1 = 0 \text{ (assumed)}$$

$$k_8 + k_9 = 150 \text{ lb/in. (assumed)}$$

$$P_3 = 500 \text{ psia (max.)}$$

The bellows area, A_8, required is a minimum of 4.95 sq in. From available sizes, a bellows area of 6.0 sq in. was selected. Using this value in Equation (A-11), actuator pressures, P_3, were plotted as a function of stroke as shown in Figure A-24. Since stroke vs flow area is a linear function (see Figure A-6) and flow area vs flow is also linear (the dump valve, PN 393140, operates in a choked condition) the curve in Figure A-24 is also proportional to the actuator pressure vs flow requirements.

Servo Controller Design (PN's 393090, 393088, 393140)

Referring to the schematics in Figures A-1, A-2, and A-4, the actuator controllers selected for use are constant bleed systems using an electrical torque motor to vary actuation pressure in response to changes in input signal.

Calculations were made to determine servo controller output pressure as a function of the torque motor wand position for various input signals. The results are shown in Figures A-25 through A-27. A sample calculation of one point is as follows. (Calculations are for PN 393090.)

Selected servo controller areas and stroke

$$A_1 = 3.14 \times 10^{-4} \text{ sq in. (0.020 in. dia)}$$

$$A_2 = 0 \text{ to } 15.9 \times 10^{-4} \text{ sq in. (0.045 in. dia)}$$

$$S_2 = 0.004 \text{ in. (selected stroke)}$$

$$\frac{S_2}{r_{S_2}} = \frac{0.004}{0.0225} = 0.178$$

$$\frac{A_2}{15.9 \times 10^{-4}} = 0.355 \text{ (from Figure A-6, wand operation is similar to a flat poppet)}$$

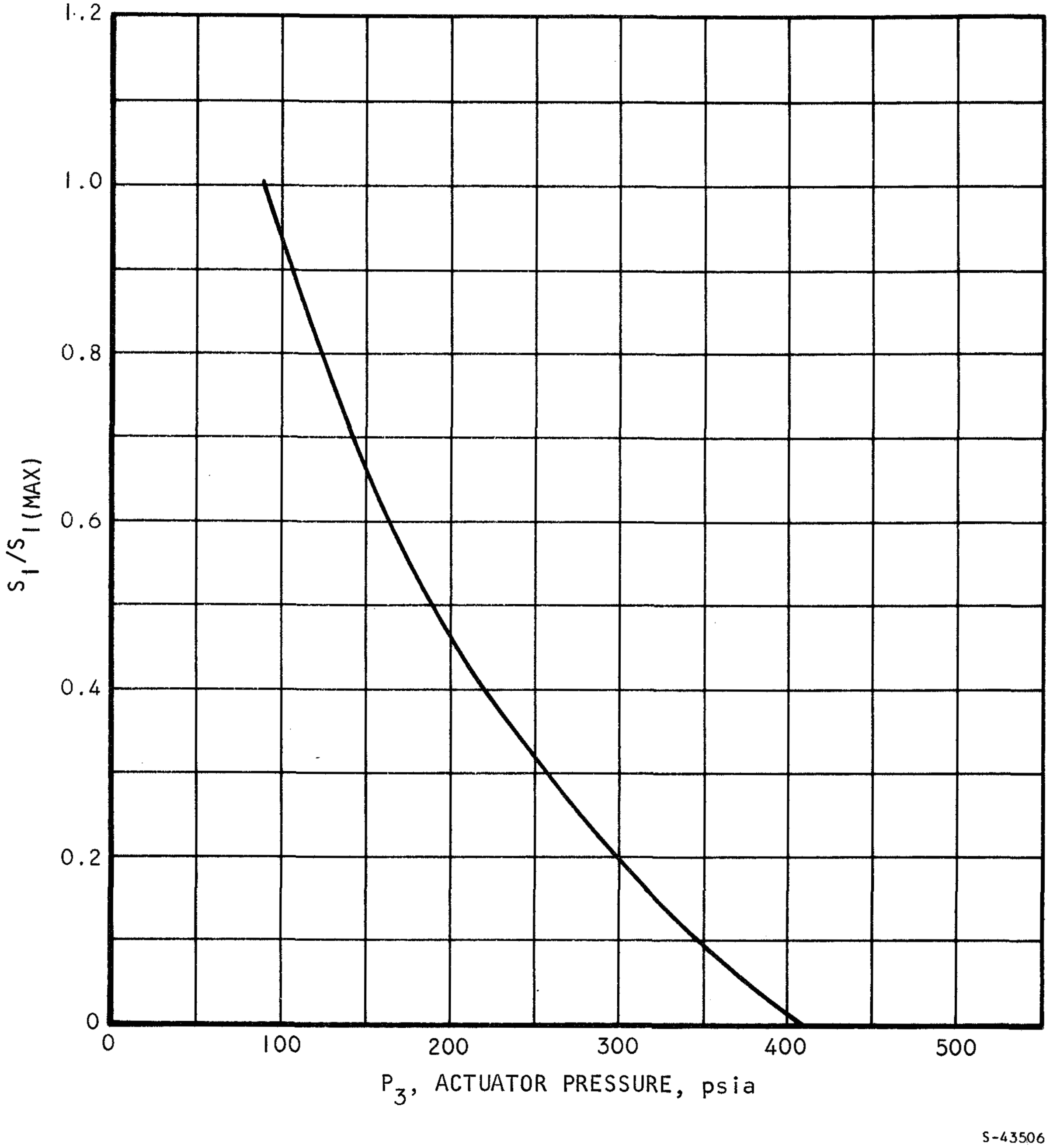


Figure A-24. Stroke Ratio vs Actuator Pressure (PN 393140)

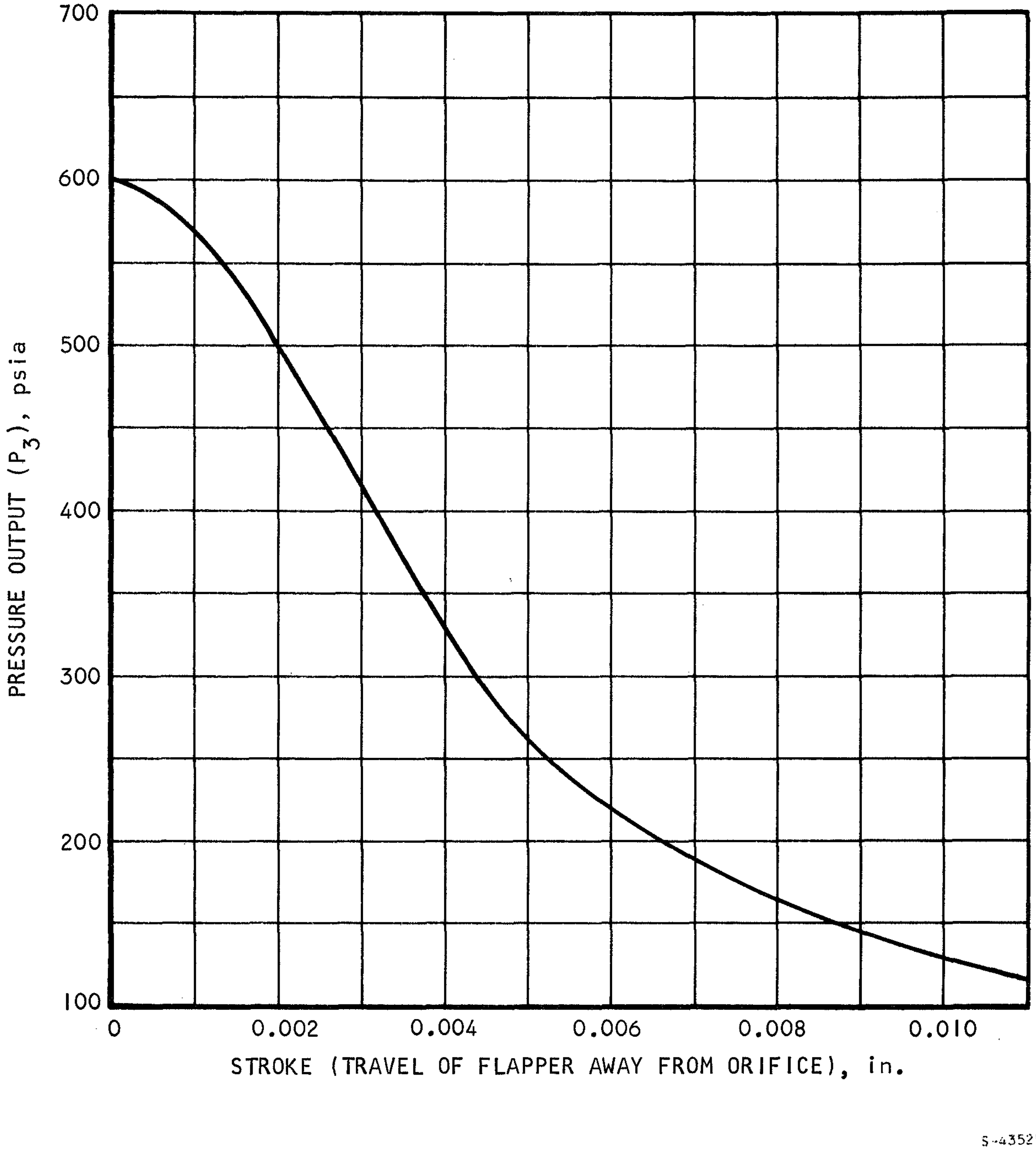


Figure A-25. Servo Output Pressure vs Torque Motor Stroke (PN 393090)

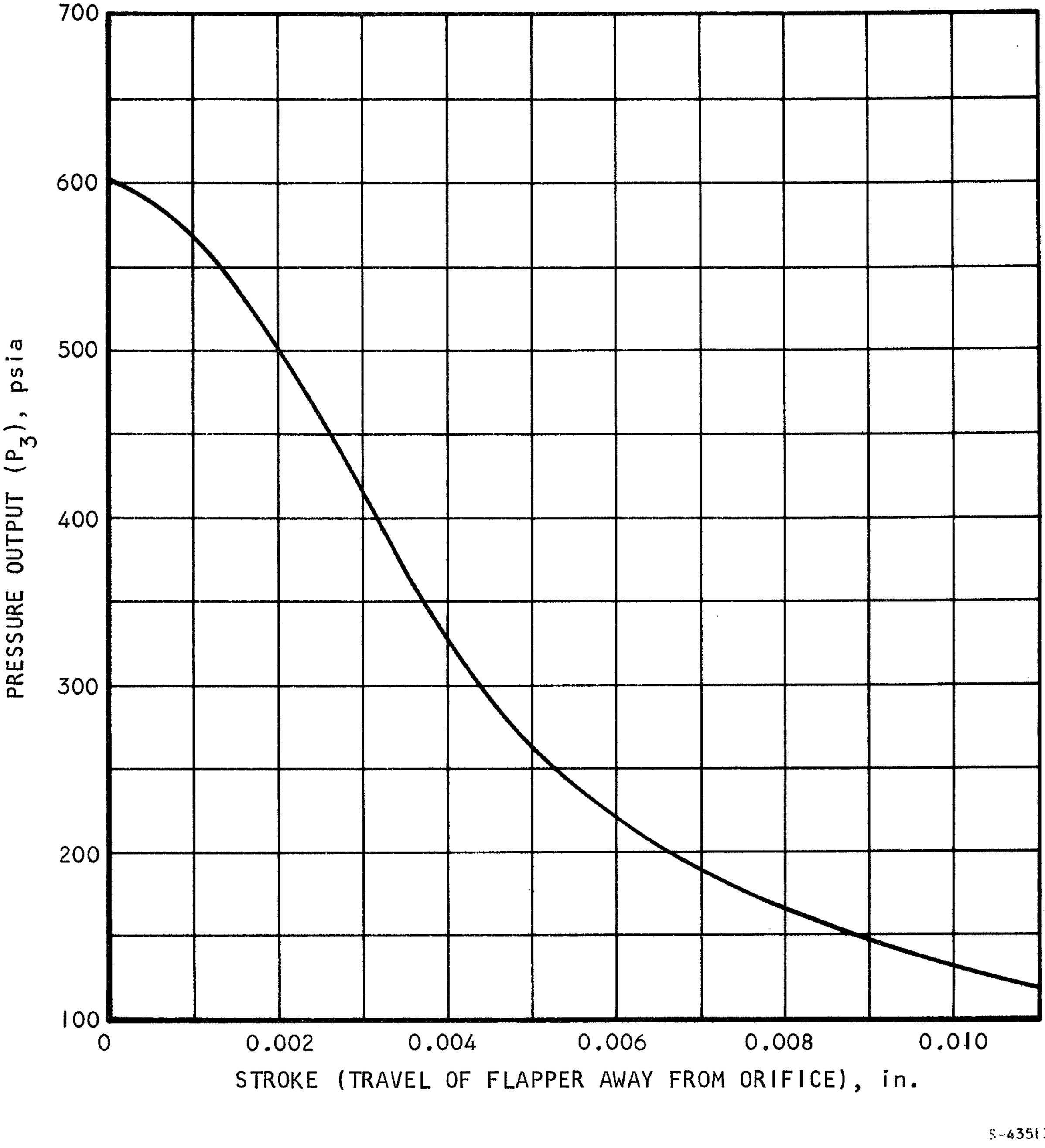


Figure A-26. Servo Output Pressure vs Torque Motor Stroke (PN 393088)

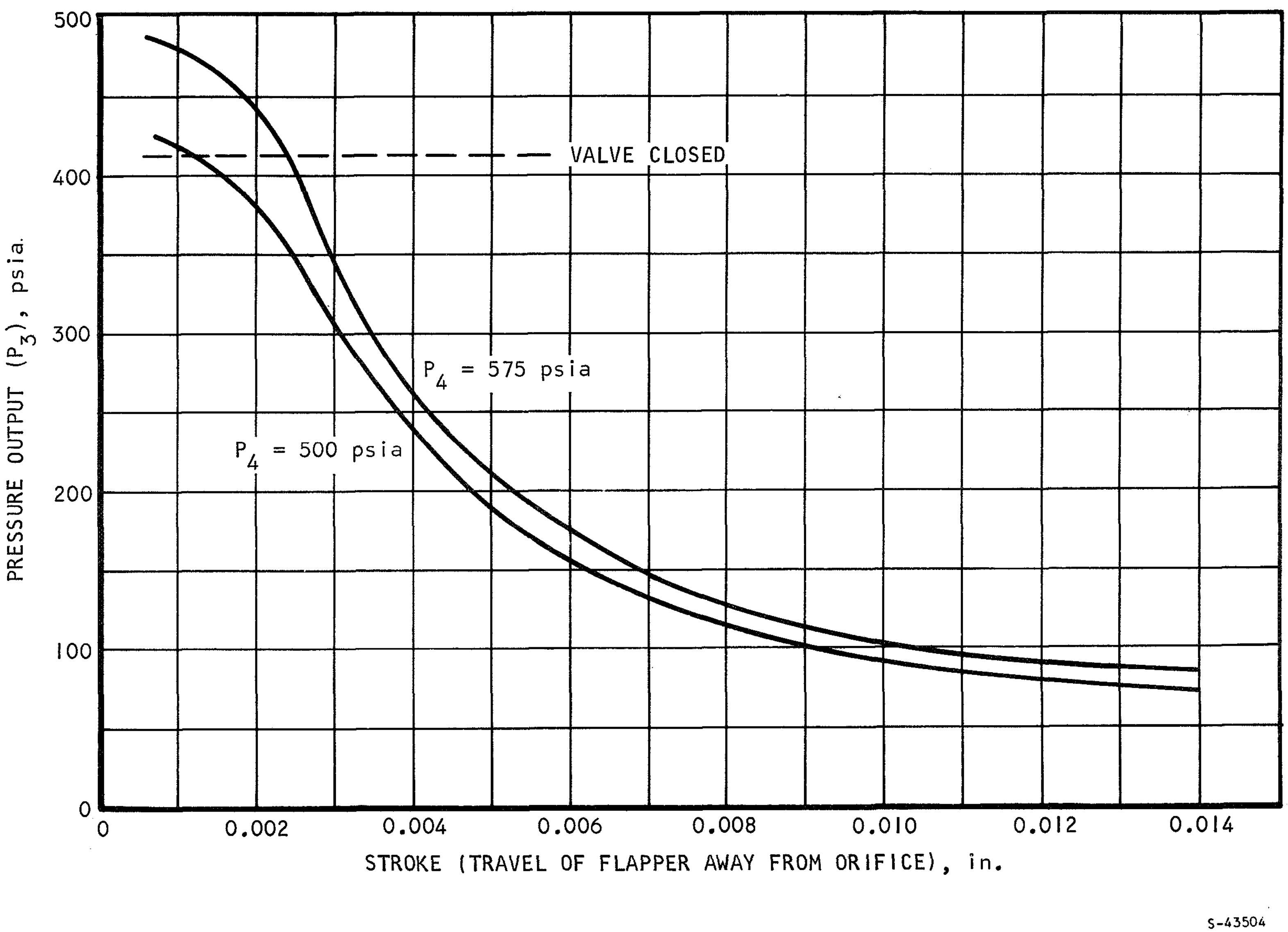


Figure A-27. Servo Output Pressure vs Torque Motor Stroke (PN 393140)

$$A_2 = (0.355)(15.9 \times 10^{-4}) = 5.64 \times 10^{-4} \text{ sq in.}$$

$$A_1/A_2 = \frac{3.14 \times 10^{-4}}{5.64 \times 10^{-4}} = 0.556$$

$$P_3/P_4 = 0.55 \text{ (from Figure A-7--} P_3/P_4 \equiv P_y/P_z\text{)}$$

$$P_3 = (0.55)(600) = 330 \text{ psia}$$

$$F_A = P_3A_8 = (330)(3.98) = 1310 \text{ lb}$$

Figure A-28 is a curve of torque motor stroke vs input current obtained from test data from the torque motor to be used.

Combining this curve with the above curves, and previous calculated flow vs actuation pressure requirements, the following flow vs torque motor input curves are developed as illustrated in Figures A-29 through A-31.

Servo Controller Design (PN 393094)

For a given downstream pressure, P_2, the actuation pressure, P_3, required to balance the poppet bellows forces, and the controller poppet flow area, A_6, can be calculated as follows:

$$P_2 = 300 \text{ psia (selected value)}$$

$$P_1 = 525 \text{ psia (nominal upstream pressure)}$$

$$A_7 = 0.0012 \text{ sq in. (0.039 in. dia seat design size selection)}$$

$$A_5 = 0.0008 \text{ sq in. (0.032 in. dia orifice design size selection)}$$

$$P_5 = P_2$$

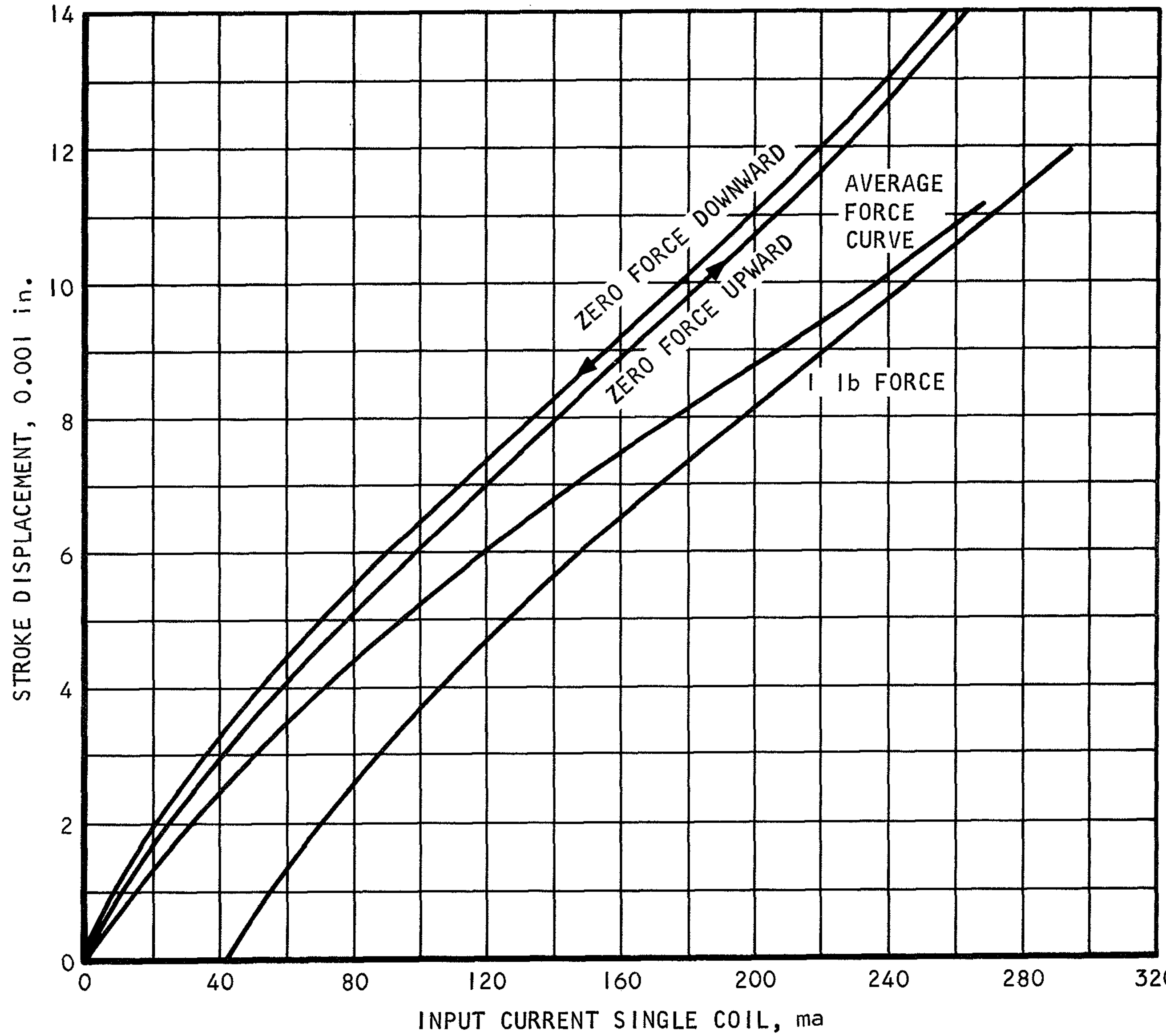
$$P_1 - P_2 = 525 - 300 = 225 \text{ psi}$$

$$P_1 - P_3 = 50 \text{ (from Figure A-23)}$$

$$P_3 = 525 - 50 = 475 \text{ psia required actuation pressure}$$

$$P_1A_5N_{1-3} = P_3A_6N_{3-5}$$

The above relationship was found by applying Equation (A-1) to the flow-through-orifice areas, A_5 and A_6, and assuming constant temperature. The N function is equal to:



S-43502

Figure A-28. Stroke vs Input Current

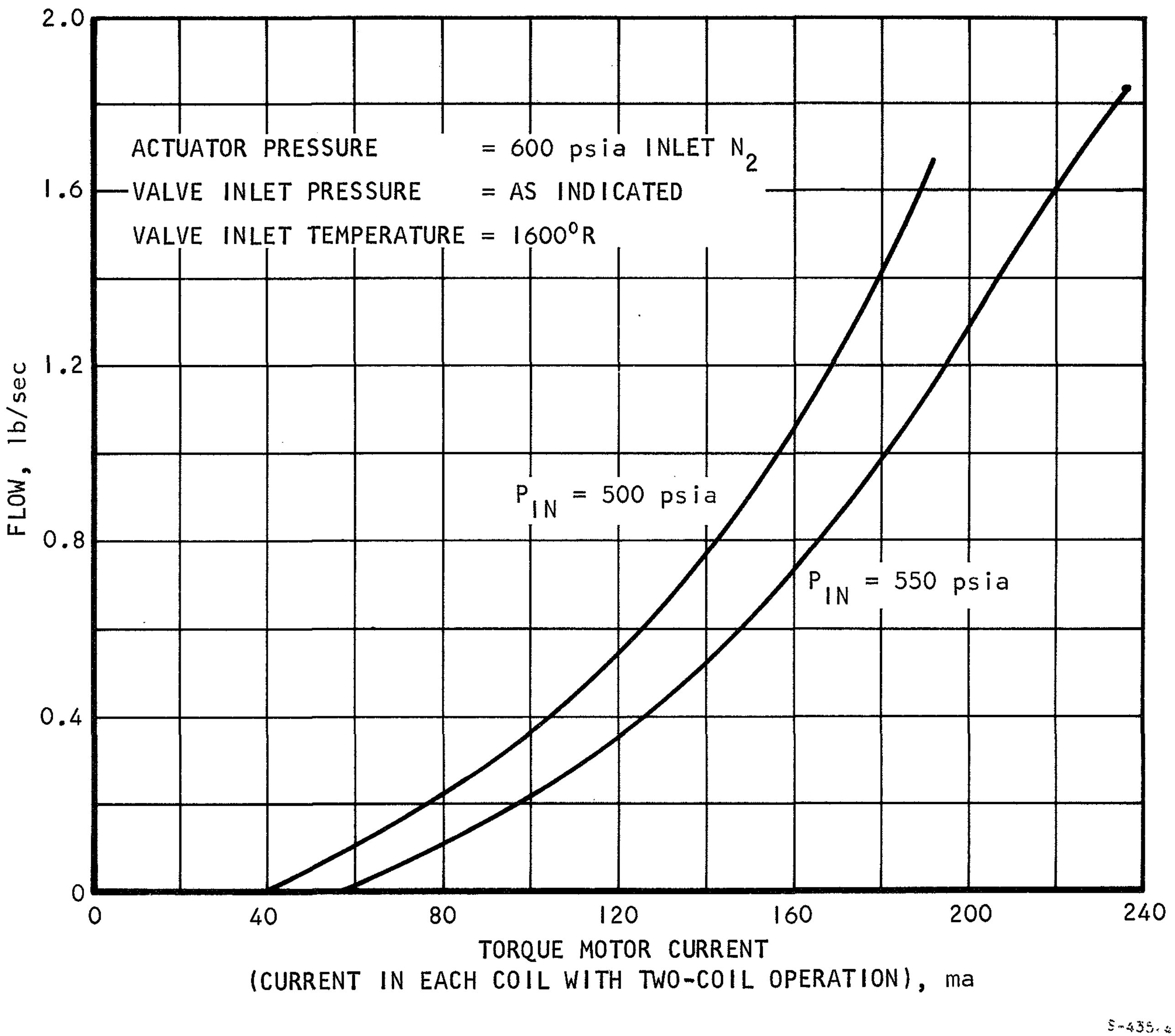


Figure A-29. Flow vs Torque Motor Current (PN 393090)

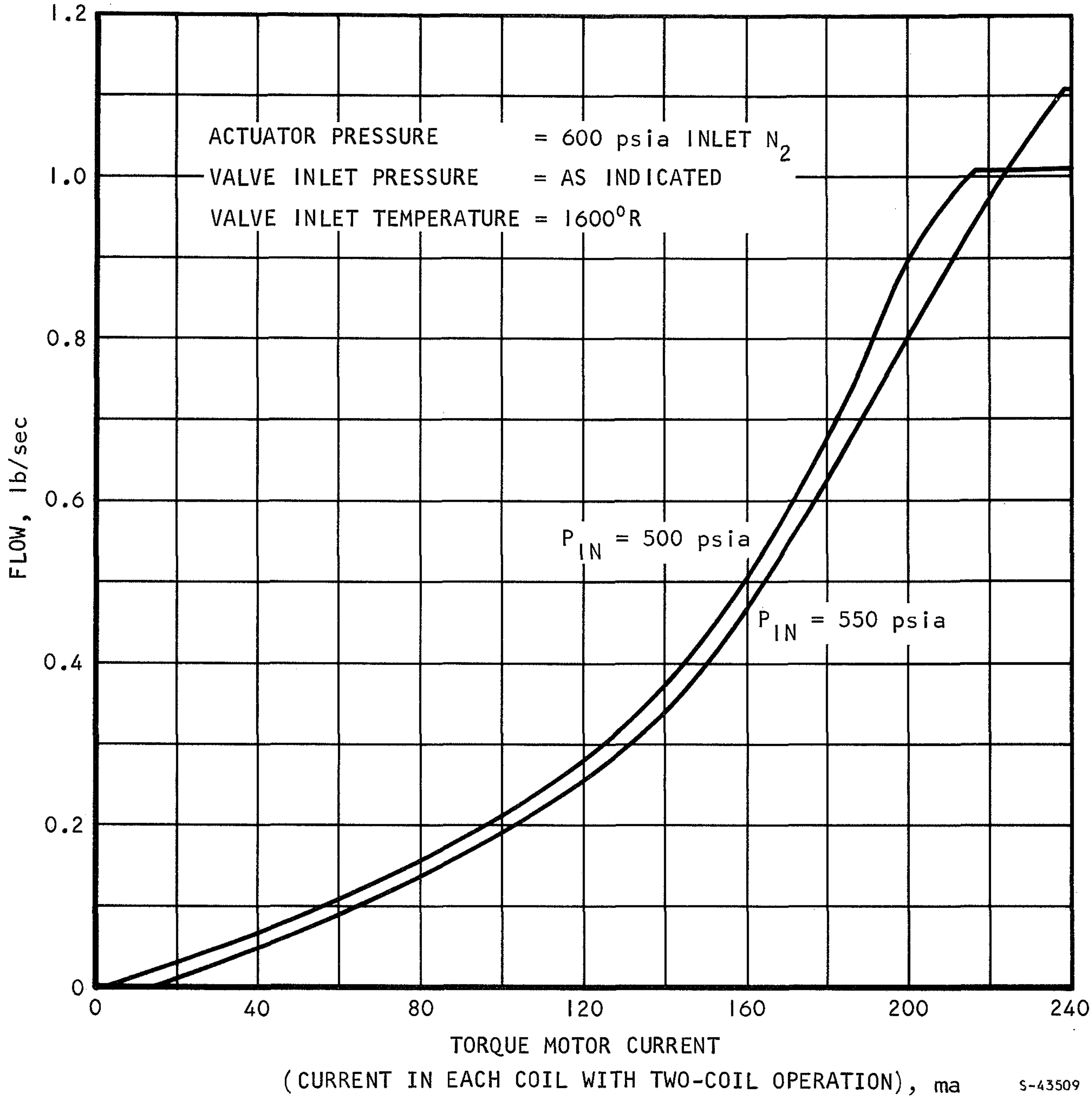
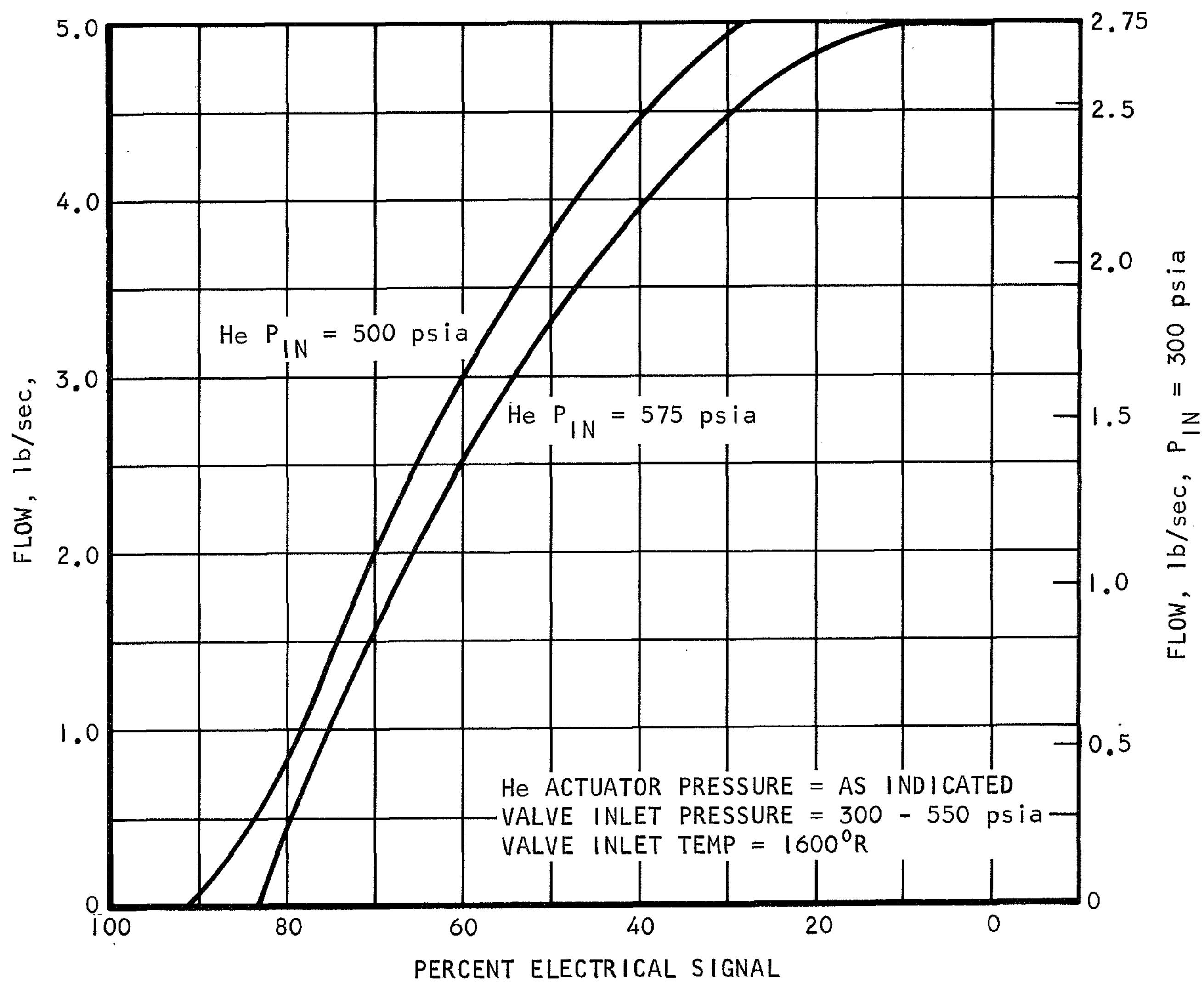


Figure A-30. Flow vs Torque Motor Current (PN 393088)



S-43512

Figure A-31. Flow vs Input Signal (393140)

$$N_{1-3} = \frac{\sqrt{\frac{\gamma}{\gamma - 1}\left[\left(\frac{P_3}{P_1}\right)^{\gamma} - \left(\frac{P_3}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}}{\left[\sqrt{\frac{\gamma}{\gamma - 1}\left[\left(\frac{P_3}{P_1}\right)^{\gamma} - \left(\frac{P_3}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right]}\right]_{max.}}$$

with $P_3/P_1 = \frac{475}{525}$, $N_{1-3} = 0.604$

and $P_5/P_3 = \frac{300}{475}$, $N_{3-5} = 0.976$

$$A_6 = \frac{(525)(0.0008)(0.604)}{(475)(0.976)} = 0.000548 \text{ sq in.}$$

Several additional values were calculated and the results are shown in Figure A-32. The maximum area ratio, A_6/A_7, is, from Figure A-32, equal to 0.885. Figure A-33 is a plot of flow area vs stroke for a spherical poppet with a 90-deg contact angle. From this figure it has been determined that the maximum controller poppet stroke is 0.0113 in.

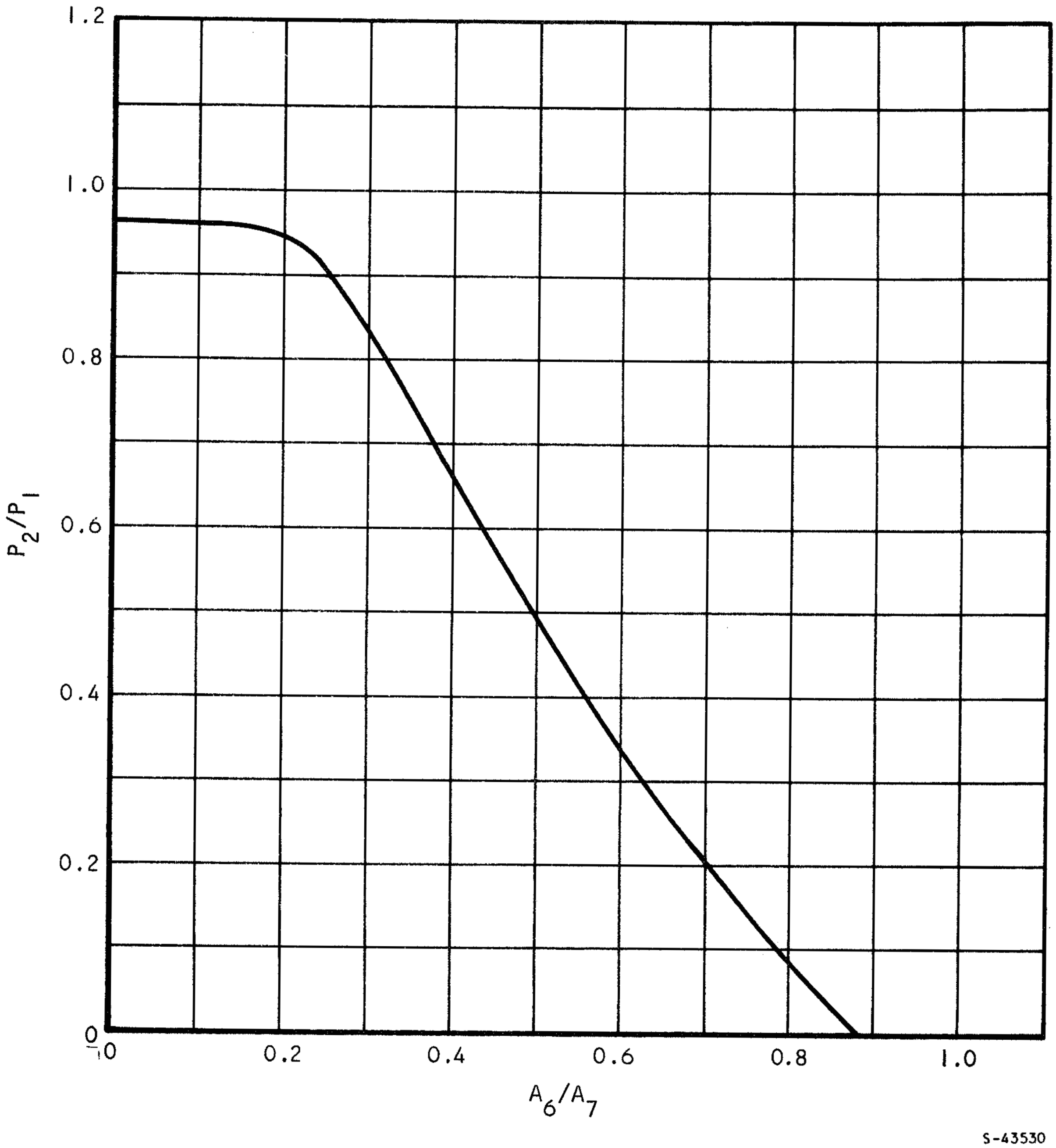


Figure A-32. Pressure Ratio vs Metering Area (PN 393094)

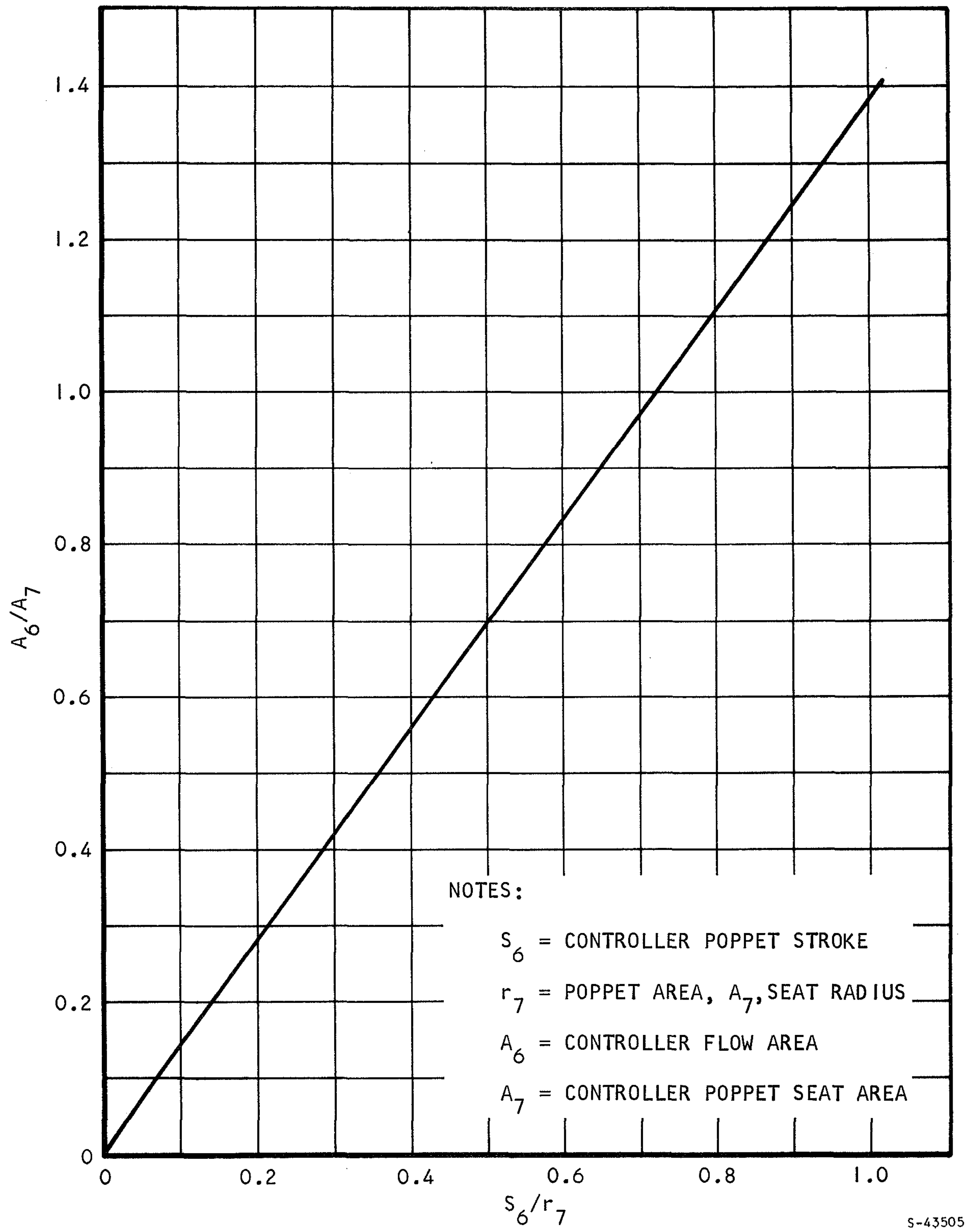


Figure A-33. Flow Area vs Poppet Stroke for 90-deg Spherical Poppets